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About Authors

● **W. E. Benninghoff** *joined the Ohio Crankshaft Co. to assist in the development of local hardening of bearing surfaces by high frequency induction, now known as the Tocco process, and is now manager of the Tocco Division. He has been active in the application of electric heat to the heat treating of steels for the past 11 years. Last Fall in collaboration with M. A. Tran, he presented a paper on "Surface Hardening by Induction" before the National Metal Congress. He has also spoken on the subject before several SAE Sections. Mr. Benninghoff received his B.S. degree in electrical engineering from the Case School of Applied Science in 1920.*

● **Frank W. Caldwell** *first began study of propeller vibration problems some 21 years ago. Since then he received particular commendation when the Collier Trophy was awarded to Hamilton Standard Propellers in 1933 in connection with the development of the controllable-pitch propeller, and was personally honored by being selected to receive the Reed Award in 1935, for his work in this field. During the official investigation of the recent Hindenburg disaster Mr. Caldwell was the expert called upon to make a study of the propellers in the wreckage. Before joining the Hamilton Standard Propeller organization in 1929 he had been affiliated with the Curtiss Aeroplane Co. and the Propeller Branch of the U. S. Army Air Corps, Materiel Division. Other papers by Mr. Caldwell have been published in earlier issues of the SAE Journal.*

● **R. M. Critchfield** *is chairman-elect of the SAE Indiana Section. A graduate from Ohio State University in 1916, a member of Eta Kappa Nu and Sigma XI—honorary electrical and scientific societies—he has devoted his work to electrical engineering. Before the War he was with Westinghouse; during it he was in the U. S. Navy, eventually assuming charge of the fire control division; after it he was assistant chief engineer with Dyneto Electric Corp., before joining the Remy Electric Division of G.M.C. in 1921. Now, as chief engineer of the Delco-Remy Division, he also finds time to fill the office of president, Anderson, Ind., Board of Education.*

(*Continued on page* 28)

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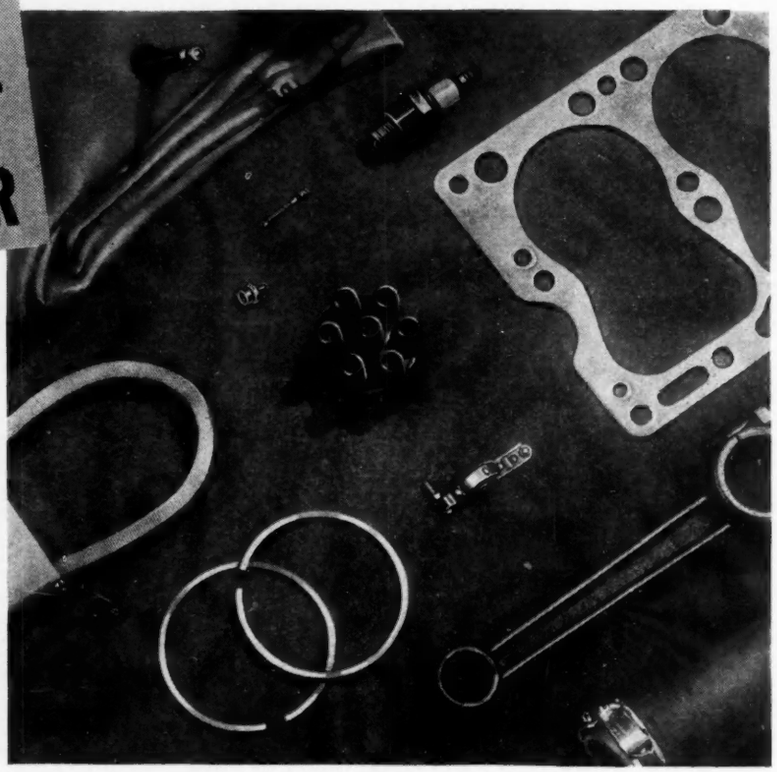

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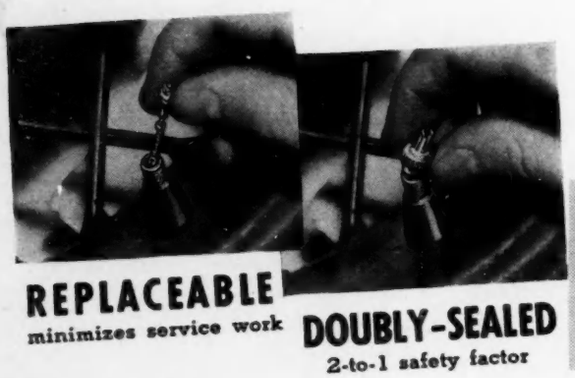

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TIRE VALVES

Crankcase-Oil Temperature Control

By Ellis W. Templin

Automotive Engineer, General Plant Division, Department of Water and Power, City of Los Angeles, Calif.

TO supplement previous information from the manufacturer's or refiner's point of view, research and operating data from the fleet operator's standpoint are presented.

The need for crankcase temperature control is quite evident during cold- and hot-weather operation. Operating conditions such as roads, grades, and atmospheric temperatures are used in connection with crankcase-oil temperatures to give data demonstrating the need of oil temperature control.

Plans are described whereby crankcase-oil temperature may be controlled to aid in easy starting at atmospheric temperatures below 70 deg. fahr. Definite proportions and capacities of temperature control units are suggested. Some are adaptable for the combined purpose of quickly raising the temperature of crankcase oil to normal operating temperature and controlling the oil temperature against excessively high operating temperatures.

THE problem of crankcase-oil temperature control has been discussed at previous S.A.E. meetings from various angles; therefore, it is only possible that in this paper we may present some ideas and data from slightly different angles, and especially from the point of view of an operator of a large fleet of cars and trucks. In presenting this paper, it is not the purpose to set forth a detailed method for complete solution of the problem, but to bring to bear upon the subject such data and information as have come to the attention of the writer in his experience with automotive operation in the Department of Water and Power, City of Los Angeles, Calif.

The purpose of controlling crankcase-oil temperature is to provide better engine lubrication and thereby to reduce engine wear, and to improve engine performance and economy.

Need of Temperature Control

A sufficient quantity of operating data is available to indicate that engine cylinder wear due to corrosion is accelerated at low operating temperatures, especially at starting. Excellent support for this statement is found in Mr. C. G. William's report on tests conducted by the Institution of Automotive Engineers of Great Britain.* When an engine and its crankcase oil are cold, the high viscosity of the oil prevents an adequate flow. This condition encourages wear by providing insufficient oil to support lubrication.

Cold conditions at starting put a heavy load upon the battery and starter. When the battery happens to be weak, the low temperature makes it still less efficient than normal. Combined, the conditions make for uncertain and unreliable starting.

After starting, if considerable time elapses in bringing the engine cooling water and crankcase oil up to normal operating temperature, considerable damage is done not only to the engine but to the mind and nerves of the operator.

When the engine has reached a satisfactory operating temperature, it is desirable to hold the crankcase-oil temperature at approximately that of the cooling water. This condition is required in order to avoid unnecessary oil "drag." When driving hard, it is desirable to hold the temperature of the crankcase oil as close as possible to that of the cooling water in order that the viscosity of the crankcase oil shall not become dangerously low. This latter condition is required in order to avoid metal-to-metal contact and consequent wear.

A volume of unpublished data from petroleum refiners and automotive-engineering sources justifies the statement that injurious chemical products of oxidation are developed by many of our better oils at some temperature above 250 deg. fahr. Etching and excessive deterioration of cadmium-silver and copper-lead bearings, also stuck rings and valve stems, have been traced definitely to the acids and gum products of high-temperature oil breakdown. Many refiners are now adding anti-oxidation elements to their oils to correct this natural weakness. These protected oils are expensive and are not generally available.

Approach to Solution

To obtain the best lubricating conditions it would be desirable to have a lubricating oil with a uniform viscosity of close to 60 sec. Saybolt Universal (S.U.) between minimum and maximum operating oil temperatures. 60 sec. S.U. viscosity is taken for the purpose of this paper as the minimum permissible crankcase-oil viscosity (on the basis of new oil at the maximum operating temperature). Some authorities believe it to be a practical safe value for providing adequate lubrication which may allow for unfavorable factors such as dilution and "lugging."

The next best condition may be obtained by the use of some means of maintaining the crankcase oil and cooling water at a reasonable temperature for easy starting. Also, by some

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 5, 1937.]

* Many bibliographical references were made use of in the preparation of this paper. A list of these references may be had upon request to S.A.E. Headquarters.

device, the crankcase oil may be brought up quickly at starting to approximately the cooling-water operating temperature. The same device would then be available for holding the crankcase-oil temperature down to approximately the temperature of the cooling water under heavy or hard-driving conditions. The functions desired to be accomplished are shown graphically in Fig. 1. In the main it represents A.S.T.M. viscosity-temperature charts for the various oils used by the Department. (Note: Oils mentioned hereafter in this paper, described by this chart, are designated "D.S.A.E. 20," "D.S.A.E. 30," and so on.) In addition, line *A-A* is drawn along the 60 sec. S.U. line as indicating the ideal uniform-viscosity lubricant. Although desirable, such an oil is not at present available on the market.

At low temperatures some feature or mechanism is desired capable of coming into play when the crankcase oil falls below 70 deg. fahr. and being of such capacity as to prevent the oil from falling below 40 deg. fahr., even though the atmospheric temperature may be as low as —30 deg. fahr. At 40 deg. fahr. the viscosity of D.S.A.E. 20 oil becomes 2,000 sec. S.U., a value that makes for easy starting, and certainly much easier than the viscosity of 200,000 sec. S.U. which would prevail if the oil temperature were —30 deg. fahr.

The desired effect on the oil viscosity by the use of such means is illustrated at *B* in Fig. 1.

A means similar to that used for crankcase-oil control should also be applied to the cooling water. The water should be maintained at or above 40 deg. fahr. to aid starting further.

Engine performance is poor and cylinder wear is greatest at cylinder temperatures below 140 deg. fahr. Hence, it is advisable to bring the engine crankcase oil and cooling water to a temperature of 170 deg. fahr. as soon as possible. The warm-up period should not exceed 2 min.

Above the normal temperature of 170 deg. fahr., danger lies in the possibility of excessive oil temperatures which may reduce the viscosity of the oil below the recommended minimum of 60 sec. S.U. and thereby cause damage to the engine. Means to prevent this increase in temperature should have the effect of maintaining a suitable viscosity of the oil as illustrated at *C* in Fig. 1.

It is recognized that lubricating oil is a very poor coolant due to its low specific heat, but it is the only medium at hand. Hence it appears that we should use it as best we may until our engine designers provide water jackets for the bearings, pistons, and valves.

The specific heat of oil is approximately one-half that of water at ordinary temperatures, as shown by Fig. 2.

With the desirable functions for controlling the crankcase-oil temperature mentioned previously, accomplished, we may be assured of easy starting and very nearly perfect lubrication at starting and properly protected bearings under heavy-driving conditions.

Viscous Flow

An illustration is given here to explain the reason that oil flows so reluctantly at low temperatures.

From Schoder & Dawson's book on Hydraulics we have the formula for viscous flow of liquids, as follows:

$$P = 0.000273 \times \frac{LVQ}{D^4}$$

where P = Loss of pressure in pounds per square inch in a tube.
L = Length of tube in feet.
V = Viscosity in centipoises (absolute).
D = Inside diameter of the tube in inches.
Q = Flow in gallons per minute.

A few values chosen and substituted will make it possible for one to visualize the effect of viscosity upon the flow.

Solving the formula for flow Q.

$$Q = \frac{PD^4}{0.000273 \times LV} \text{ or } \frac{PD^4}{0.000273 \times L} \times \frac{1}{V}$$

Assuming a ⅜-in. O.D. copper tube, which is 0.311 in. inside diameter, an oil pressure of 40 lb. per sq. in., a length of tube of 5 ft., and viscosities (Department S.A.E. 60 oil):

At 270 deg. fahr. 60 sec. Saybolt, equivalent to 9.5 centipoises.

At 210 deg. fahr. 108 sec. Saybolt, equivalent to 20 centipoises.

At 70 deg. fahr. 5,500 sec. Saybolt, equivalent to 1,100 centipoises.

At 40 deg. fahr. 20,000 sec. Saybolt, equivalent to 4,000 centipoises.

At 0 deg. fahr. 300,000 sec. Saybolt, equivalent to 60,000 centipoises.

Note: Viscosities in centipoises may be determined by reference to Fig. 3. A line drawn from the S.U. value (on the Kinematic-Saybolt Scale to the left) to the specific gravity of the oil (about 0.92) crosses the absolute viscosity line at a point indicating the equivalent value in centipoises. It will be observed that, above 200 sec. S.U., the relation of absolute viscosity in centipoises to S.U. seconds is in the ratio of 1:5, below that it varies as shown by Fig. 4.

Substituting the values just given in the formula, we have:

$$Q \text{ at } 270 \text{ deg. fahr.} = \frac{40 \times 0.311^4}{0.000273 \times 5} \times \frac{1}{9.5} = 28.900 \text{ gal. per min.}$$

$$Q \text{ at } 210 \text{ deg. fahr.} = 274 \times \frac{1}{20} = 13.700 \text{ gal. per min.}$$

$$Q \text{ at } 70 \text{ deg. fahr.} = 274 \times \frac{1}{1,100} = 0.249 \text{ gal. per min.}$$

$$Q \text{ at } 40 \text{ deg. fahr.} = 274 \times \frac{1}{4,000} = 0.0685 \text{ gal. per min.}$$

$$Q \text{ at } 0 \text{ deg. fahr.} = 274 \times \frac{1}{60,000} = 0.00457 \text{ gal. per min., or approximately 0.6 fluid oz. per min.}$$

These figures indicate the important influence that temperatures below 70 deg. fahr. have upon the ability of oil to be delivered to the bearing surfaces, especially when one realizes that these calculations were based upon a ⅜ in. outside diameter tube, whereas many oil passages in an engine are much smaller.

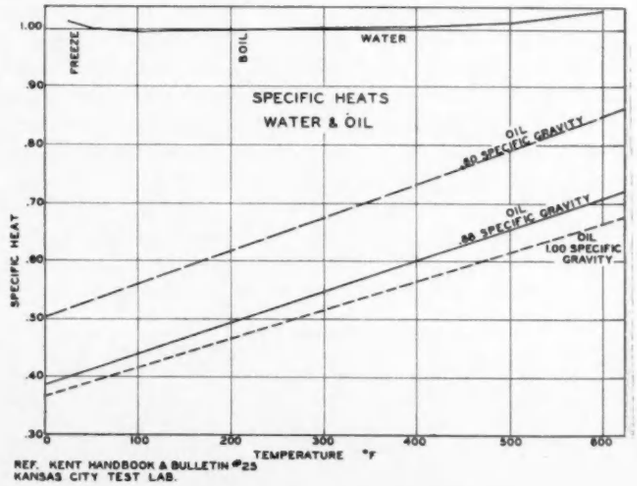


Fig. 2 – Specific Heats of Oil and Water

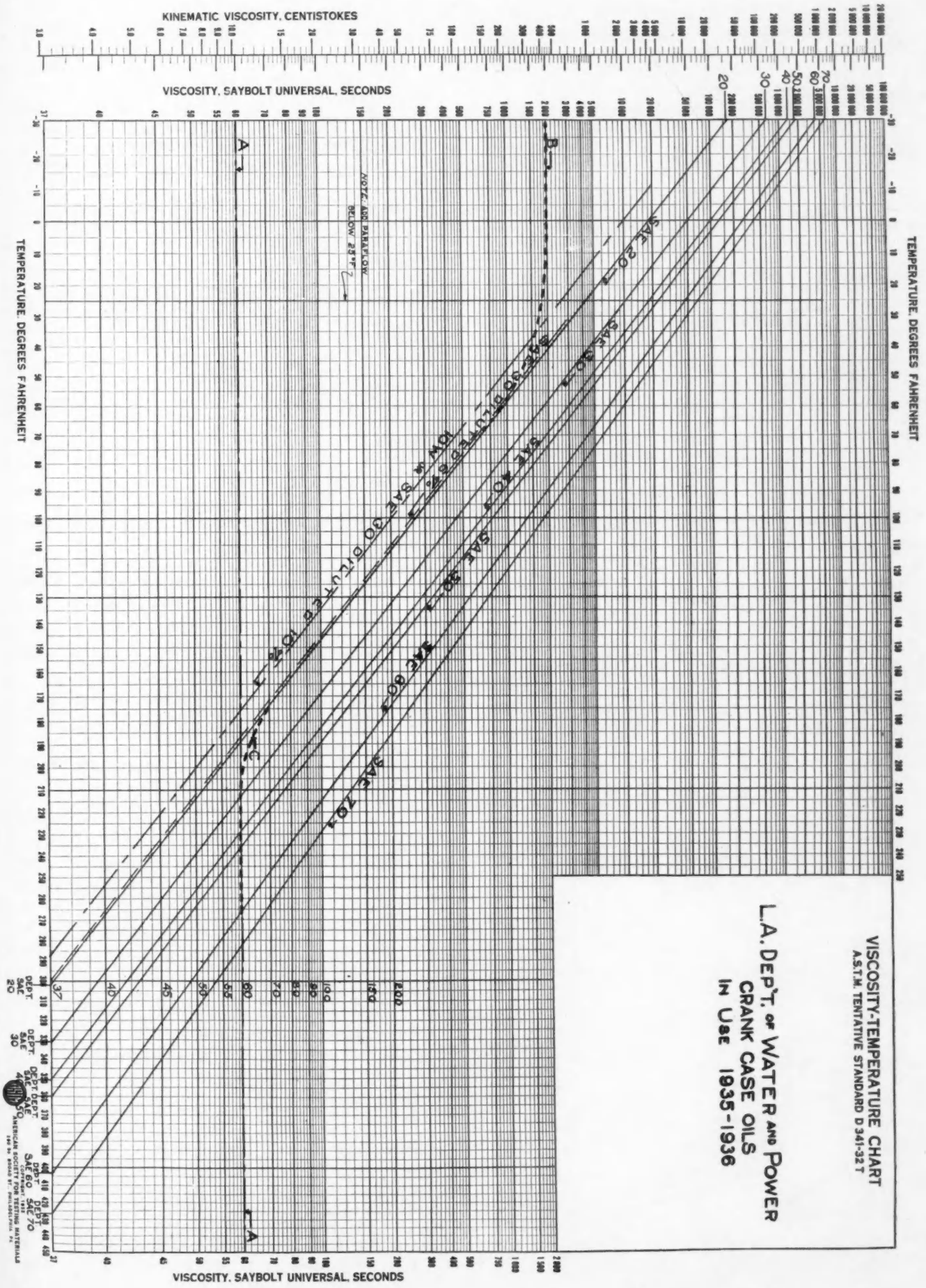


Fig. 1 – Los Angeles Department of Water and Power Crankcase Oils

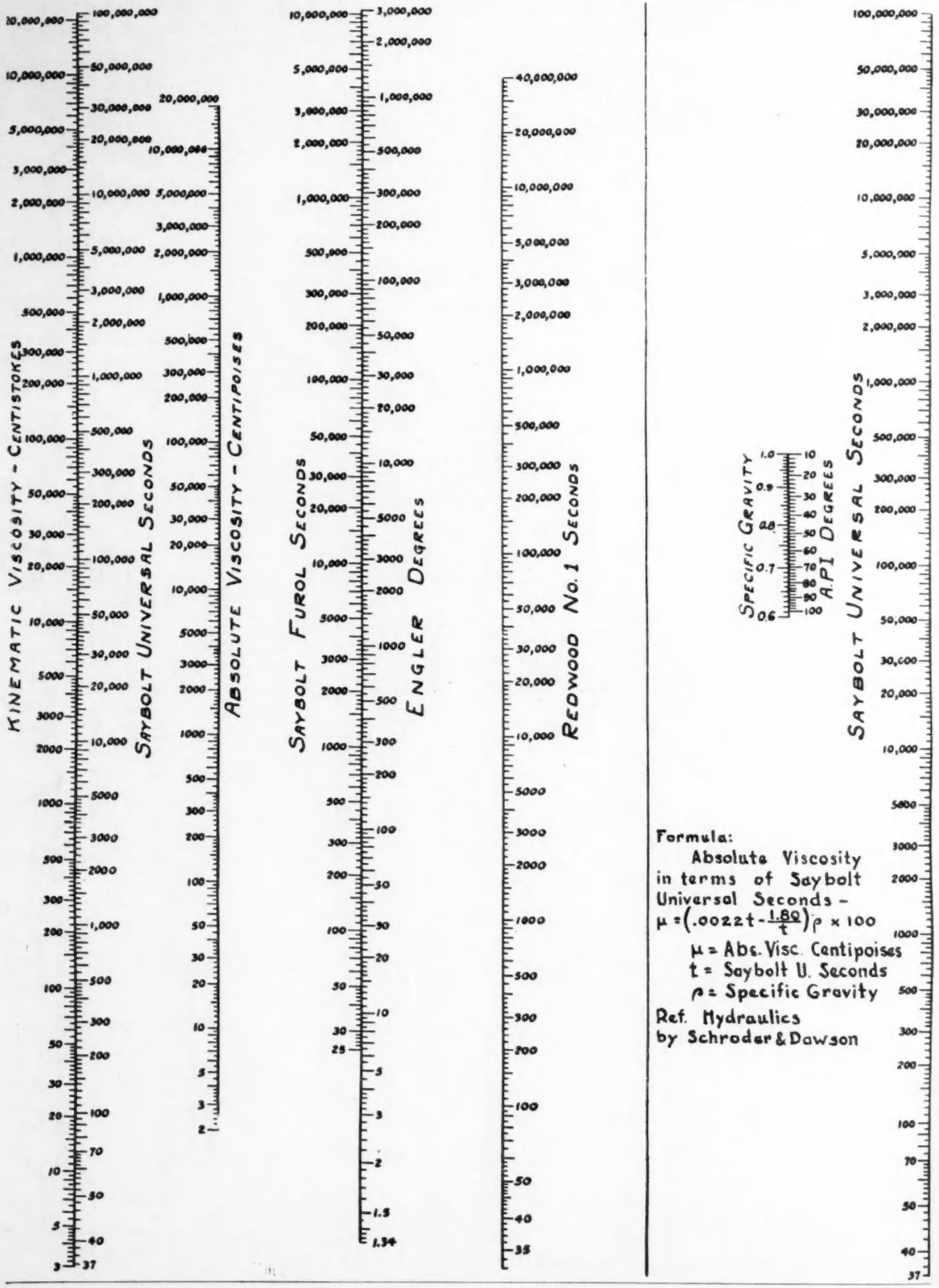
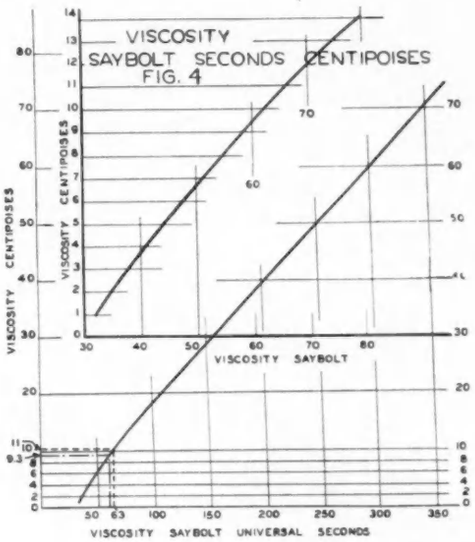

KINEMATIC VISCOSITY - CENTISTOKES
SAYBOLT UNIVERSAL SECONDS
ABSOLUTE VISCOSITY - CENTIPOISES
SAYBOLT FUROL SECONDS
ENGLER DEGREES
REDWOOD No. 1 SECONDS
SPECIFIC GRAVITY
A.P.I. DEGREES
SAYBOLT UNIVERSAL SECONDS
Formula:
Absolute Viscosity in terms of Saybolt Universal Seconds -
$\mu = \left(.0022t - \frac{1.80}{t}\right)\rho \times 100$
μ = Abs. Visc. Centipoises
t = Saybolt U. Seconds
ρ = Specific Gravity
Ref. Hydraulics by Schroder & Dawson

Fig. 3 – Viscosity Conversion Chart



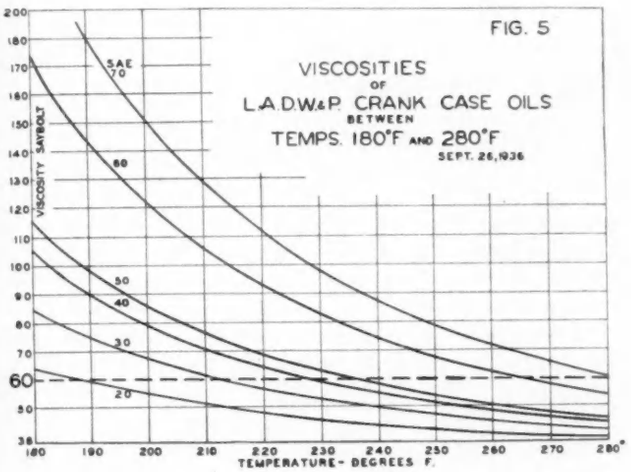


Fig. 4 – Viscosity Saybolt Seconds to Centipoises (Absolute)

Fig. 5 – Viscosities of Los Angeles Department of Water and Power Crankcase Oils at High Temperatures

High Temperatures

It is known definitely that excessively high crankcase temperatures will reduce the oil viscosity below the safe limit with resultant engine damage. Operating informations indicate that good lubrication will prevail if conditions do not reduce the viscosity of crankcase oil below 60 sec. S.U., assuming, of course, that other factors are normal.

Fig. 5 shows the viscosities of Department oils in the high-temperature range. In this diagram an enlarged scale for viscosities is taken to "magnify" their values with reference to temperatures. Occasionally diagrams of this kind are plotted so as to indicate that, at high temperatures, the viscosities of all oils appear very nearly the same. However, this diagram shows definitely that there are material differences at the hazardous temperatures. For instance, if the crankcase temperature does not exceed 190 deg. fahr., D.S.A.E. 20 oil is safe. If the temperature reached 200 deg. fahr., it is doubtful whether D.S.A.E. 20 oil would be safe. There is a spread of 20 deg. fahr. between D.S.A.E. 20 oil and D.S.A.E. 30 oil at the hazardous temperatures that should be given serious consideration. A similar spread occurs between D.S.A.E. 50 and D.S.A.E. 60 oils. Failure to take these temperature differences into account may result in excessive maintenance costs.

Operating experience and consideration of internal engine temperatures lead to the conclusion that crankcase-oil temperatures should not exceed 210 deg. fahr.

Operating Conditions

Let us examine operating conditions with reference to a certain fleet of cars and trucks. We will refer particularly to the fleet operated by the Department and information gained therefrom. This fleet is composed of 1256 motor-vehicle units, made up of 430 passenger cars and 826 trucks. The truck units vary from light-duty "pickups" to 60,000-lb. gross capacity tractor semi-trailers and 68,000-lb. gross capacity trucks with trailers. Last year this fleet operated over 15,000,000 vehicle miles.

Basic operation is carried on within the city limits of Los Angeles. However, major construction projects and operating service route our vehicles across the desert, between Los Angeles and Boulder Dam (300 miles to the east) and along the Los Angeles Aqueduct into the High Sierras (350 miles to the north). See Figs. 6 and 7. The operating elevations vary from 200 ft. below sea level to 8000 ft. above. Atmospheric temperatures vary from —26 deg. fahr. to 130 deg. fahr. Road conditions include congested city traffic, paved high-speed highways, hundreds of miles of dirt and sand road, sub-zero winter snow, high-temperature desert, and long mountain grades.

A very severe run is described by reference to Figs. 7, 8, and 9. Fig. 7 shows the Los Angeles Aqueduct and Mono Basin Extension, with the state highway nearby, going from Los Angeles through San Fernando, Palmdale, Lancaster, Mojave, Little Lake, Independence and Bishop to Mono Lake,

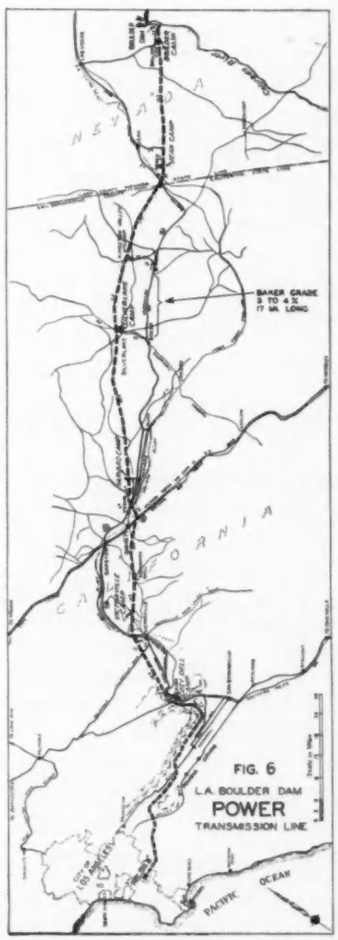


Fig. 6 – Los Angeles Boulder Dam Power Transmission Line

Fig. 7 – Los Angeles Aqueduct – Mono Basin Extension

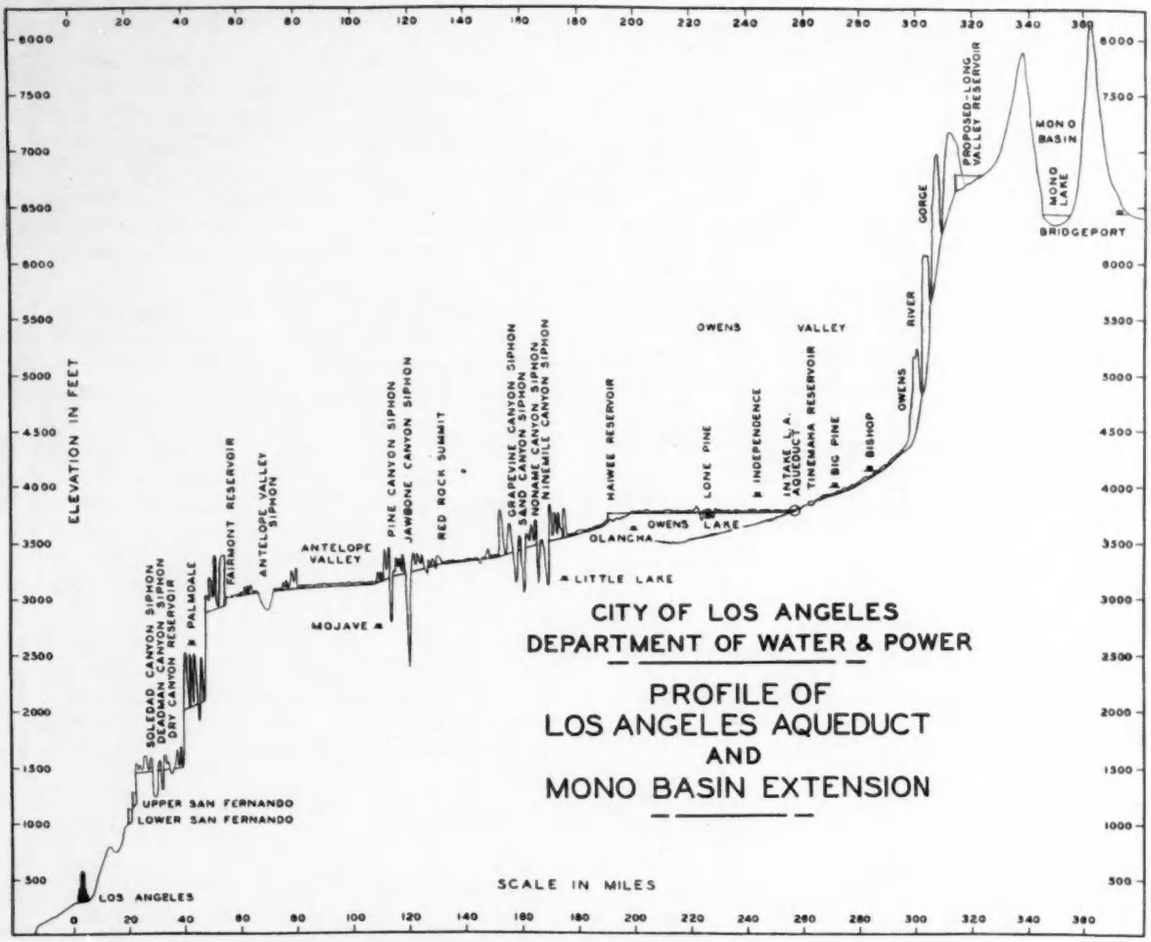


Fig. 8 – Profile Map – Los Angeles Aqueduct and Mono Basin Extension

Calif. (350 miles north of Los Angeles). Fig. 8 shows the variation in elevation from Los Angeles (300 ft.) to the High Sierras near Mono Lake (reaching above 8000 ft.). Fig. 9 is a grade chart showing the per cent of grades and their lengths as plotted from information secured from the drawings on file in the California State Highway Engineer's office. The height of the black space above and below the horizontal reference line represents the inclination of the grade in per cent. Above the reference line, reading from left to right, is represented the up-grade while going north; below the line the black spaces represent down-grade. The reverse is true on the return trip, reading from right to left.

It will be noted that Sherwin grade is the longest steep grade on the run. It is approximately 8 miles long and most of it is between 7 and 8 per cent. Operation on this grade represents the most severe condition of the entire run. At Ricordo (or Red Rock Canyon) sometimes an unfavorable desert tailwind will produce crankcase temperatures very nearly equal to those on Sherwin.

The Department has a few vehicles permanently equipped with accurate distance-type thermometers with bulbs installed in the crankcase about 1 in. from the "floor." Drivers take crankcase-temperature readings while on actual service runs. Where special studies are desired, a portable 10-point potentiometer is used.

Atmospheric Temperatures

U. S. Weather Bureau and Department thermometers have shown the atmospheric temperatures throughout the Department's system, representative of those prevailing on the Mono Run, as given in Table I. They are maximum and minimum, by months, as obtained during the past 5 years.

Table II gives the maximum observed crankcase temperatures on a series of runs made by a 12,000-lb. gross capacity chassis equipped with a 20-passenger bus body. This bus

TABLE I

Los Angeles Department of Water and Power System

Atmospheric Temperatures Maximum and Minimum by Months

	L.A. Aqueduct Past 5 Years			Boulder Trans.	Power Line
	Min	Max		Min	Max
January	-26°F	98°F	------	24°F	70°F
February	-26°	94°	------	34.5°	77.8°
March	-12°	103°	------	42°	95°
April	9°	97°	------	48°	105°
May	12°	104°	------	53°	115°
June	24°	108°	------	54°	118.5°
July	20°	118°	------	69.8°	130°
August	24°	109°	------	80°	119°
September	14°	109°	------	64°	112°
October	0°	105°	------	34.8°	99°
November	-15°	97°	------	33.8°	79.5°
December	-22°	88°	------	23.5°	72°

TABLE II

Observed and Estimated Crankcase Temperatures

20 Passenger Bus on Los Angeles - Mono Run.

Date	Gross Weight	Location	Observed Atmos. Temp. Deg. F.	Max. Atmos. Temp. Deg. F.	Diff. Bet. Observed Temp. & Max. Temp.	Cooling Water Temp. Deg. F.	Max. Observed CC Temp. Deg. F.	(a) DSAE Visc. No. Oil	(b) 3/4 Atmos. Temp. Diff.	(c) Est. Max. CC Oil Temp.	(d) DSAE Visc. Oil
4- 7-36	20 Pass. Bus	Sherwin Grade	60	104	44		185	20	33	218	40
4-16-36	"	Sherwin Grade	80	104	24		200	30	18	218	40
4- 9-36	"	Sherwin Grade	55	104	49		185	20	37	222	40
4-10-36	"	Mojave	86	116	30		194	20	22	216	30
4-11-36	"	Sherwin Grade	65	104	39		195	20	30	225	40
4-23-36	"	Red Rock Canyon	84	118	34	195	200	30	26	226	40
4-23-36	"	Sherwin Grade	65	104	39	180	195	20	30	225	40
4-25-36	"	Red Rock Canyon	76	118	42	190	195	20	31	226	40
4-25-36	"	Sherwin Grade	60	104	44	180	190	20	33	223	40
4-25-36	"	Homestead	85	118	33	180	198	30	25	223	40
5-14-36	"	Red Rock Canyon	96	118	22		210	30	17	227	40
2-18-37	"	Mint Canyon	57	115	58	190	222	40	44	266	60
2-18-37	"	Red Rock Canyon	56	118	62	180	219	40	47	266	60
2-18-37	"	Sherwin Grade	34	104	70	180	204	30	53	257	60
2-19-37	"	Independence	54	106	52	200	200	30	39	239	50
2-19-37	"	Homestead	59	118	59	190	240	50	44	284	70
3- 2-37	"	Red Rock Canyon	67	118	51	175	210	30	38	248	60
3- 3-37	"	Homestead	73	118	45	180	216	30	34	250	60

Note: "CC" represents crankcase

operates regularly between Los Angeles and the Mono Basin. Column (*a*) gives the D.S.A.E. viscosity of oil dictated by the observed maximum crankcase temperature and selected by reference to Fig. 5 (60 sec. S.U. as a minimum).

Column (*b*) is ¾ the difference between the observed atmospheric temperature and the maximum reported at or near the point of observation.

Column (*c*) represents anticipated or estimated maximum crankcase temperature, assuming that the car were operating at the location given when maximum atmospheric temperature prevailed. The crankcase temperature would be increased by approximately ¾ the difference between the observed temperature and the maximum. This is an average allowance, usually on the safe side.

Column (*d*) represents the D.S.A.E. viscosity of oil dictated by the estimated maximum crankcase temperature selected from Fig. 5 as before.

According to observed crankcase temperatures, grades of oil from D.S.A.E. 20 to 50 are indicated as proper. Estimated maximums indicate as a rule D.S.A.E. 40 oil, and in some cases 60 and even 70.

D.S.A.E. 40 oil is used in this engine at all times and appears quite successful, as may be seen by the following shop record:

Rebored at 74,099 miles – traveled 45,614 miles since (still going).

"Re-ringed" at average of 21,341 miles.

Rod bearings replaced at average of 28,077 miles.

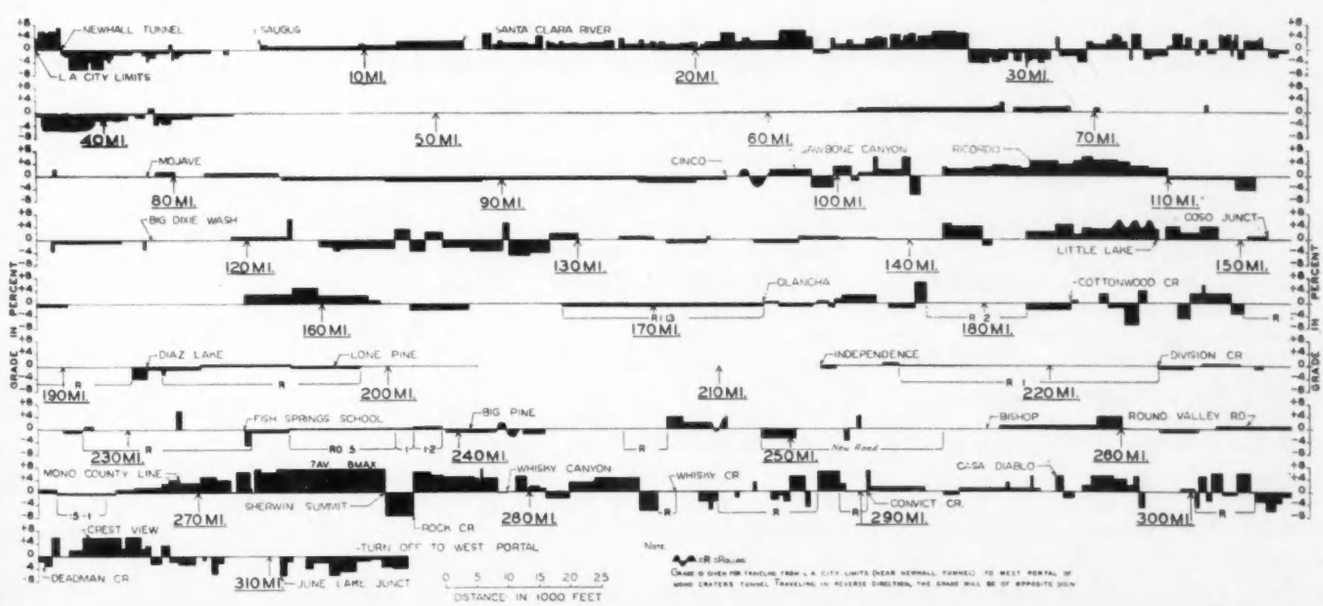


Fig. 9 – Grade Chart – Newhall to West Portal

TABLE III

Observed and Estimated Crankcase Temperatures

10-Ton Truck with 140 H.P., 6 Cylinder, 5 x 5-3/4 Engine
Los Angeles - Mono Run

Date	Gross Weight	Location	Observed Atmos. Temp. Deg. F.	Max. Atmos. Temp. Deg.F.	Diff.Bet. Observed Temp. & Max.Temp.	Cooling Water Temp. Deg. F.	Max.Observed CC Temp. Deg. F.	(a) DSAE Visc. No. Oil	(b) 3/4 Atmos. Temp. Diff.	(c) Est.Max. CC Oil Temp.	(d) DSAE Visc. Oil
5-25-36	34,000	Sherwin Grade	85	104	19	180	235	50	15	250	60
5-26-36	34,000	Sherwin Grade	95	104	9	170	240	50	7	247	60
5-28-36	68,000	Sherwin Grade	70	104	34	170	240	50	25	265	60
6-12-36	68,000	Mint Canyon	105	115	10	200	255	60	7	262	60
9-18-36	68,000	Mint Canyon	100	115	15	170	225	40	11	236	50
9-18-36	68,000	Sherwin Grade	75	104	29	160	190	20	22	212	30
9-24-36	68,000	Red Rock Canyon	115	118	3	190	230	40	2	232	50
9-24-36	68,000	Sherwin Grade	60	104	44	160	175	20	33	208	30
10- 1-36	68,000	Mint Canyon	92	115	23	165	220	40	17	237	50
10- 1-36	68,000	Sherwin Grade	75	104	29	150	215	30	22	237	50
10- 6-36	68,000	Mint Canyon	85	115	30	160	210	30	22	232	40
10- 6-36	68,000	Sherwin Grade	90	104	14	180	210	30	11	221	40
10-15-36	68,000	Sherwin Grade	70	104	34	180	200	30	25	225	40
10-23-36	68,000	Halloran	70	130	60	170	215	30	45	260	60
10-23-36	68,000	Cajon Pass	75	116	41	175	205	30	34	239	50
12-15-36	68,000	Red Rock Canyon	45	118	73	160	195	20	57	252	60

Note: "CC" represents crankcase

In Table III, for a 10-ton truck with a 140-hp. six-cylinder, 5 by 5¾ in. engine, observed crankcase temperatures indicate the proper grades of oil to be from D.S.A.E. 20 to 60. Estimated maximums indicate 30 to 60, with 50 and 60 predominating. D.S.A.E. 60 oil is being used in this truck engine at all times, and that is the grade recommended by the engine manufacturer.

Table IV is for another 10-ton truck with a 140-hp. six-cylinder engine, but with 5 by 6 in. bore and stroke. Observed temperatures on this engine indicate D.S.A.E. 30 and 40 oil to be proper, whereas consideration for maximum atmospheric temperatures indicates 40 to 60 oil. D.S.A.E. 50 oil is being used in this engine at present, but may be changed to 60 for summer operation.

The engine manufacturer recommends D.S.A.E. 30 oil for this job.

It should be obvious that, if means were available for controlling the crankcase-oil temperature so that it would not exceed 210 deg. fahr., D.S.A.E. 30 oil (62 sec. S.U. at 210 deg. fahr.) could be used safely in the engines reported under Tables II, III, and IV.

The practical advantages would be: easier starting, lower frictional loss in the engine, with a consequent increase of power, and saving of fuel.

Would it not be fitting to standardize on one S.A.E. viscosity of oil to be used throughout the year for a complete range of heavy trucks?

As a basic consideration of the reason for lower friction loss, reference may be made to Fig. 10 which is a representative Navy bearing test showing the tendency of the lower viscosities of oil to cause lower bearing temperatures. Also, the absolute viscosity of D.S.A.E. 30 oil at 210 deg. fahr. is 10

TABLE IV

Observed and Estimated Crankcase Temperatures

10-Ton Truck with a 140 H.P., 6 Cylinder 5 x 6 Engine
Los Angeles - Mono Run

Date	Gross Weight	Location	Observed Atmos. Temp. Deg. F.	Max. Atmos. Temp. Deg.F.	Diff.Bet. Observed Temp. & Max.Temp.	Cooling Water Temp. Deg. F.	Max.Observed CC Temp. Deg. F.	(a) DSAE Visc. No. Oil	(b) 3/4 Atmos. Temp. Diff.	(c) Est.Max. CC Oil Temperature	(d) DSAE Visc. Oil
1- 5-37	34,000	Mint Canyon	58	115	57	180	205	30	43	248	60
1- 5-37	34,000	Sherwin Grade	29	104	75	160	200	30	57	257	60
1-11-37	34,000	Mint Canyon	40	115	75	180	198	30	57	255	60
1-11-37	34,000	Sherwin Grade	29	104	35	170	202	30	26	228	40
2-16-37	34,000	Victorville	66	114	48	180	200	30	36	236	50
2-19-37	68,000	Little Lake	88	107	19	175	218	40	14	232	40
2-19-37	34,000	Sherwin Grade	34	104	70	160	210	30	53	263	60

Note: "CC" represents crankcase

centipoises, whereas that of D.S.A.E. 60 oil is 20 centipoises. Fluid friction is proportional to absolute viscosity, hence the heavy oil develops just twice the friction of the lighter oil.

Other advantages accrue, including the lower tendency of the lighter oil to form "gummy" deposits behind the rings and on valve stems.

Various Engine Temperatures

In conjunction with the operating temperatures reported previously, it may be of interest to refer to Fig. 11 which shows representative operating temperatures within the automotive engine in two sets of figures, one set for moderate driving of 20 to 40 m.p.h., and the other for hard driving at 40 to 70 m.p.h.

The oil viscosities under the influence of the various temperature conditions are shown in Fig. 12.

Let us examine the crankcase-oil temperature as related to connecting-rod bearing temperature. In the hard-driving range, the temperature shown ranges from 220 deg. fahr. to 350 deg. fahr. (average 285 deg. fahr.) at the connecting-rod bearing, and 170 deg. fahr. to 260 deg. fahr. (average 215 deg. fahr.) for the crankcase oil. The difference between the averages is 70 deg. fahr. From this comparison it may be concluded that, in the heavy-driving range, the bearing temperature runs approximately 70 deg. fahr. higher than the crankcase-oil temperature.

Adding this figure to the previously specified maximum desirable crankcase oil temperature (210 deg. fahr.) gives 280

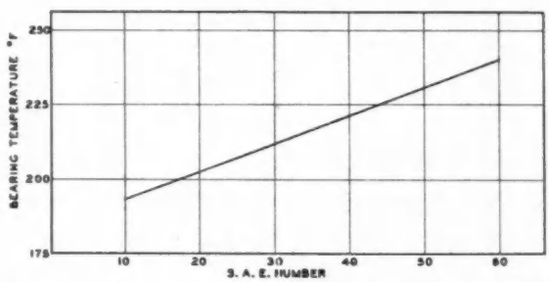


Fig. 10 – Effect of Oil Grade on Bearing Temperature

deg. fahr. as the corresponding connecting-rod bearing temperature. This is a very satisfactory maximum bearing temperature for effecting long life in connecting-rod bearings. However, it is in the range of possible chemical change mentioned previously as taking place in some oils as low as 250 deg. fahr.

As the oil passes through the bearing exposed to this temperature, there is a tendency towards film rupture. A. W. Burwell of the Alox Corp., Niagara Falls, N. Y., in various reports of tests on 27 commonly used oils, shows that their rupture temperature varies from 280 deg. fahr. to 390 deg. fahr., or an average of 328 deg. fahr. Factors of safety for these oils in terms of degrees temperature with crankcase oil at 210 deg. fahr. and bearing temperature at 280 deg. fahr., vary from 0 to 110 deg. fahr., or an average of 48 deg. fahr.

Conclusions from Data

From the foregoing data and reasoning, the following conclusions may be drawn:

(1) There are good reasons for controlling the crankcase-oil temperature, in favor of cold starting, to not less than 40 deg. fahr. (against −30 deg. fahr. atmospheric temperature) and, at the same time, controlling the cooling-water temperature by the same means.

(2) Crankcase-oil cooling means should be provided capable of controlling the top temperature of the crankcase oil at a maximum of 210 deg. fahr. under hard-driving conditions. The same means should be capable of quickly elevating the

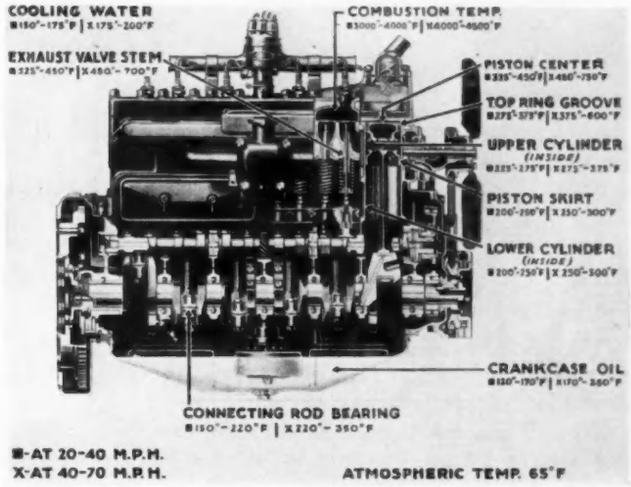


Fig. 11 – Approximate Engine Operating Temperatures

crankcase oil from starting temperature to running temperature.

How Grade of Oil is Selected

Assuming suitable crankcase-oil temperature-control means to be available, let us see how the operator's lubrication problem works out in the definite case of a representative automobile.

The representative automobile selected is one of a well-known make of the medium-price class and one of several that the Department operates. On short notice this car is expected to go anywhere on the system within a day's time.

The owner's manual (issued by the manufacturer) for this car reads as follows with respect to selection of oil for the engine:

"In the summer months, oil should be selected on the basis of the anticipated average temperature during which it is to be used.

"During the fall and winter months, an oil should be selected which agrees with the lowest temperature *likely to be encountered*. Otherwise, if too heavy oil is used, cranking and starting difficulties may be encountered when the temperature drops.

"The grade of oil best suited for various air temperatures is shown below."

S.A.E. 10W + 10 per cent kerosene	−10 deg. fahr. and colder
S.A.E. 10W	−10 deg. fahr. to +45 deg. fahr.
S.A.E. 20W	+10 deg. fahr. to +80 deg. fahr.
S.A.E. 20	+30 deg. fahr. to +80 deg. fahr.
S.A.E. 30	+50 deg. fahr. to +100 deg. fahr. and over

In connection with the selection of oils, reference also may be made to W. S. James' paper, entitled "How to Select a Motor Oil from the Standpoint of the Consumer," read before the A.S.T.M. in Chicago March 3, 1937. In this paper he presents an excellent discussion of the factors involved in such a question. In conclusion, however, he states that it is his opinion that the best way to select a motor oil from the standpoint of the consumer is for him to consult with the representatives of oil and car manufacturers and follow their advice.

In the case at hand, we, the consumer, now have the advice of the car manufacturer as quoted above. What shall we do about it?

First we shall break up the car manufacturer's recommendation into two classifications, light oils and heavy oils, and their corresponding atmospheric temperatures, as follows:

	Classification A Atmospheric Temperatures for Light Oils	Classification B Atmospheric Temperatures for Heavier Oils
S.A.E 10W + 10 per cent kerosene	−30 to +20 deg. fahr.	−30 to −10 deg. fahr.
S.A.E. 10W	+20 to +45 deg. fahr.	−10 to +10 deg. fahr.
S.A.E. 20W	+45 to +80 deg. fahr.	+10 to +30 deg. fahr.
S.A.E. 20	.....	+30 to +50 deg. fahr.
S.A.E. 30	+80 deg. fahr. up	+50 deg. fahr. up

The consumer may choose either Classification *A* or Classification *B* from this schedule and be within the manufacturer's instructions.

We will now use these instructions in connection with the Department system operating conditions. The instructions would apply to our representative automobile about as shown in Table V;

Here is the ridiculous situation where an individual consumer would have to put light oil in the engine at night so that he could start easily in the morning and then, later in the day, change to heavy oil.

For the Department the condition is not quite so bad, but still it is a problem. The practice is to put D.S.A.E. 30 oil in the crankcase of this car at Los Angeles and, when it arrives in Independence, to change to D.S.A.E. 10W for Winter operation at Mono Basin and vicinity. On the return trip the D.S.A.E. 10W is replaced with D.S.A.E. 30. The oil expense is small compared to the inconvenience and engine damage that would accrue otherwise.

This example of a serious attempt to follow the car manufacturer's instructions in actual practice presents a graphic picture of the present problem due to the lack of suitable crankcase-oil temperature control.

In thick-film lubrication the basic formula is $\frac{ZN}{P}$, where Z is viscosity of the oil in centipoises (absolute), N the R.P.M., and P the bearing pressure in pounds per square inch.

The amount of oil "drag" is proportional to Z, with N and P constant, and is important in starting.

Conversely at running, Z represents the ability of the oil to support the bearing load.

A viscosity of 60 sec. (S.U.) mentioned heretofore as a recommended minimum is equivalent to 9.5 centipoises absolute viscosity.

With these considerations in mind we may examine Table VI where is compared the oil viscosities in the same engine *with* and *without* crankcase temperature control.

TABLE V

Grade of Oil to be Used for an Automobile

(As suggested by Manufacturers Lubrication Schedule)

Month	Atmos. Temperature Min. Deg. F.	Atmos. Temperature Max. Deg. F.	Grade for Starting	Grade for Hard Driving
January	−26	98	SAE 10W + 10% Kerosene	SAE 30
February	−26	94	"	"
March	−12	103	"	"
April	14	97	" or 10W	"
May	12	104	" or 20W	"
June	24	108	SAE 10W or 20 W	"
July	20	118	SAE 10W + 10% Kerosene or 20W	"
August	24	109	SAE 10W or 20W	"
September	14	109	SAE 10W + 10% Kerosene or 20W	"
October	0	105	" "	"
November	−15	97	"	"
December	−22	88	"	"

TABLE VI

COMPARISON OF OIL VISCOSITIES WITH AND WITHOUT CRANKCASE TEMPERATURE CONTROL

WITHOUT C.C. TEMP. CONTROL

At Starting (−26° F.)

DSAE Oil	Viscosity Centipoises (Absolute)
10W	10,000 (Starting Possible)
30	80,000 (Starting Impossible)

At High Atmos. Temp. - Running

DSAE Oil	C.C.Temp. at Min. (a)Recommended Viscosity	Visc. Centipoises at Max. C.C.Temp.	Visc. % of Min. (a) Recommended Visc. Centipoises
10W	170°	(240°) 4	42% (Unsafe)
30	212°	(250°) 5.1	54% (Unsafe)

WITH C.C. TEMP. CONTROL

Crankcase Temp. Controlled to 40° Min.

DSAE Oil	Viscosity Centipoises (Absolute)
20	400 (Easy Starting)
30	900 (Easy Starting)

Crankcase Temp. Controlled to 210° Max.

DSAE Oil	C.C.Temp. at Min. (a)Recommended Viscosity	Visc. Centipoises at Max. C.C.Temp.	Visc. % of Min. (a)Recommended Viscosity Centipoises
(b)20	190°	(210°) 7	73.7%
30		(210°) 10	105% (Safe)

(a) = Minimum recommended viscosity 9.5 centipoises (Abs.).

(b) = Permissible in case bearings are fitted to approx. 3/4 thousandths inch per inch diameter of shaft for clearance or under moderate driving conditions.

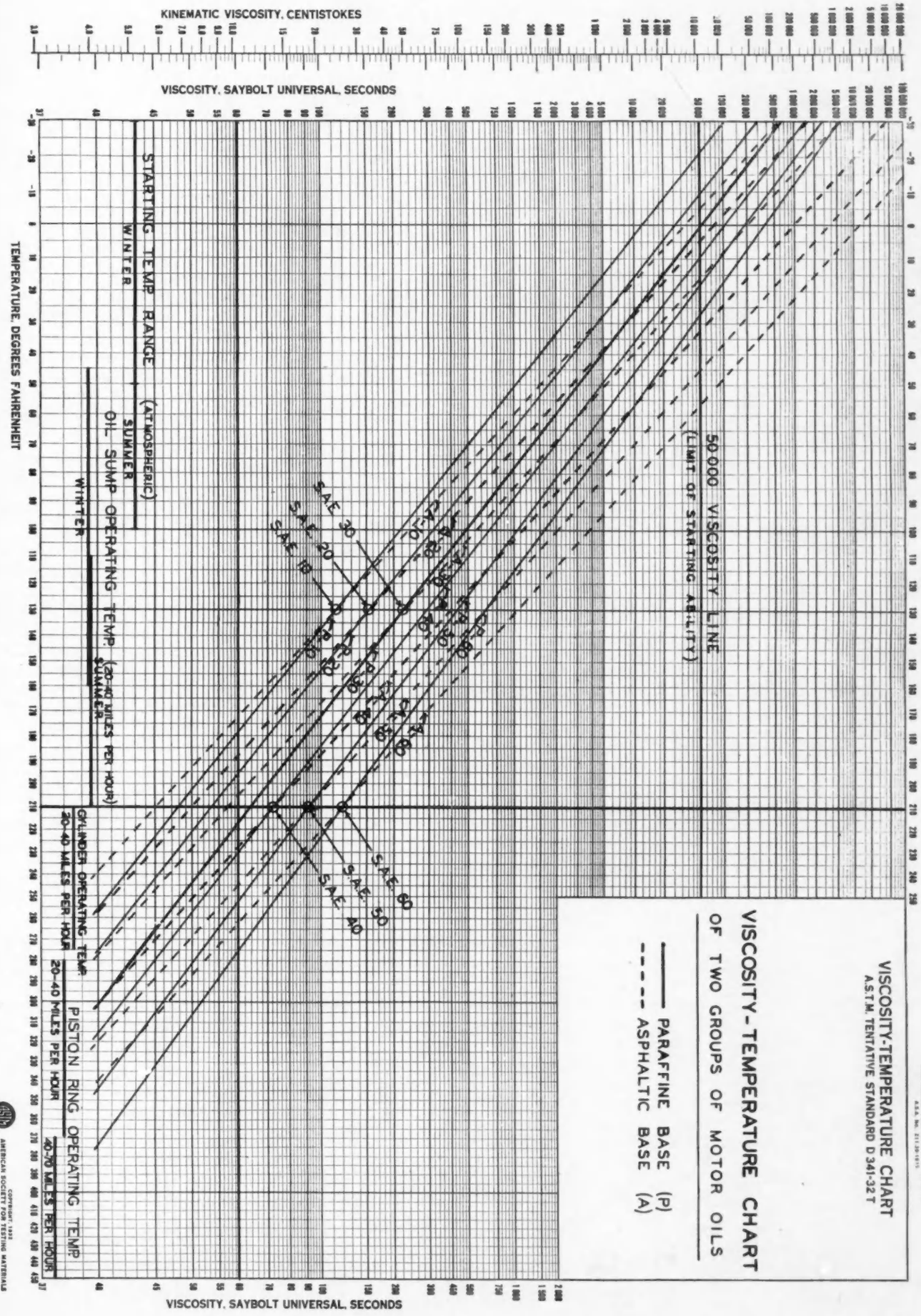


Fig. 12 – Viscosity-Temperature Chart – Two Groups of Oils

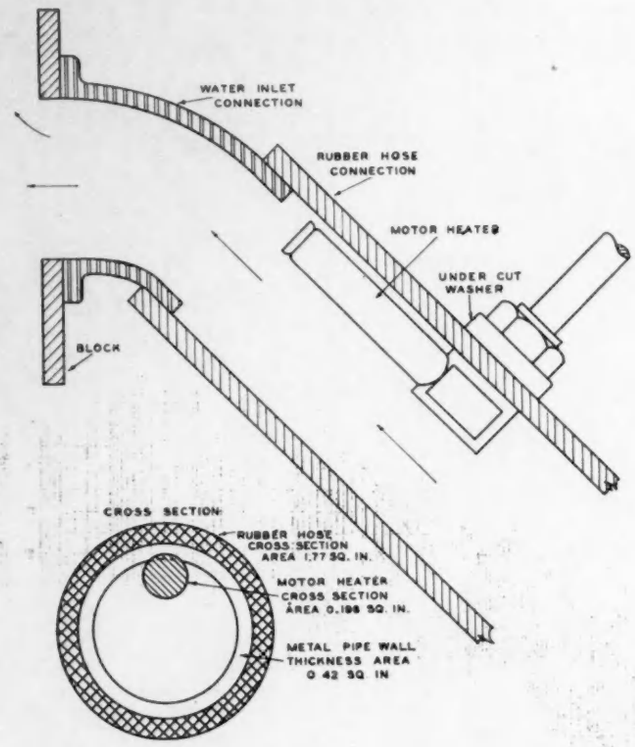

WATER INLET CONNECTION
RUBBER HOSE CONNECTION
MOTOR HEATER
UNDER CUT WASHER
BLOCK
CROSS SECTION
RUBBER HOSE CROSS SECTION AREA 1.77 SQ. IN.
MOTOR HEATER CROSS SECTION AREA 0.196 SQ. IN.
METAL PIPE WALL THICKNESS AREA 0.42 SQ. IN.

Fig. 13 – Section Showing Water Heater in Place

The conclusion is reached that, with suitable crankcase temperature control, D.S.A.E. 20 oil could be used safely throughout the year under all atmospheric conditions and still provide much easier starting than 10W oil without crankcase temperature control. This conclusion is true only for closely fitted engines or conditions of moderate driving. If D.S.A.E. 30 oil is used, there are no reservations as to bearing fits and driving conditions.

Certainly this possibility of a very simple answer to the selection of a grade of oil for continuous use deserves the careful consideration of all automotive engineers concerned.

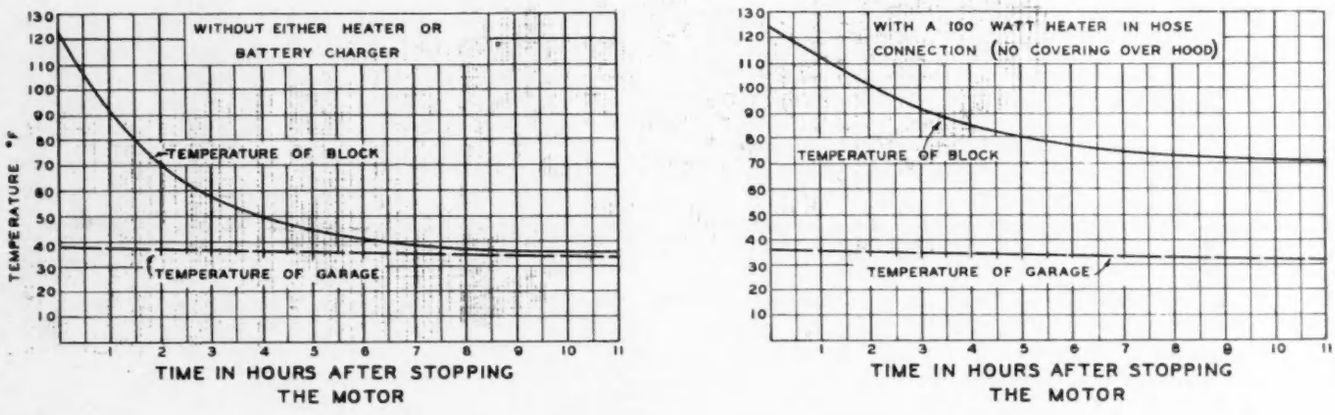
Heating Means to Aid Cold Starting

Having pointed out the reasons for controlling crankcase oil temperature and having given some of the data indicating the needed capacity of means used for that purpose, we will now review some of the work that has been done in an effort to control crankcase oil temperature.

Means for aiding cold starting from the standpoint of crankcase temperature have not been provided generally due, no doubt, to the difficulty involved in securing heat or energy from an external source. The use of an external source of heat is opposed to the basic idea of the automobile, which is an individual and self-contained transportation unit.

It might be restated here that it is not the purpose of this paper to present a complete and final answer to the problem of crankcase temperature control, but to bring such facts and data to bear upon the problem as have come to the writer's attention.

A notable effort to provide a heating element for heating the cooling water from the garage lighting circuit is described in the *General Electric Review* of December, 1932, by G. G.


WITHOUT EITHER HEATER OR BATTERY CHARGER
TEMPERATURE OF BLOCK
TEMPERATURE OF GARAGE
TEMPERATURE °F
TIME IN HOURS AFTER STOPPING THE MOTOR
WITH A 100 WATT HEATER IN HOSE CONNECTION (NO COVERING OVER HOOD)
TEMPERATURE OF BLOCK
TEMPERATURE OF GARAGE
TIME IN HOURS AFTER STOPPING THE MOTOR

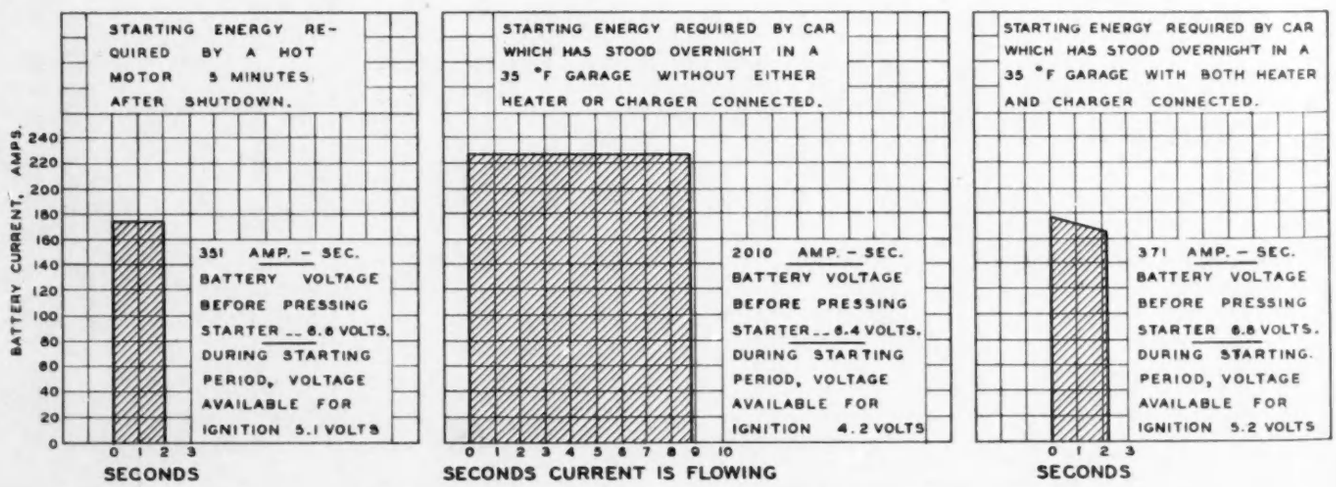

STARTING ENERGY REQUIRED BY A HOT MOTOR 5 MINUTES AFTER SHUTDOWN.
351 AMP. – SEC.
BATTERY VOLTAGE BEFORE PRESSING STARTER 6.6 VOLTS.
DURING STARTING PERIOD, VOLTAGE AVAILABLE FOR IGNITION 5.1 VOLTS
BATTERY CURRENT, AMPS.
SECONDS
STARTING ENERGY REQUIRED BY CAR WHICH HAS STOOD OVERNIGHT IN A 35 °F GARAGE WITHOUT EITHER HEATER OR CHARGER CONNECTED.
2010 AMP. – SEC.
BATTERY VOLTAGE BEFORE PRESSING STARTER 6.4 VOLTS.
DURING STARTING PERIOD, VOLTAGE AVAILABLE FOR IGNITION 4.2 VOLTS
SECONDS CURRENT IS FLOWING
STARTING ENERGY REQUIRED BY CAR WHICH HAS STOOD OVERNIGHT IN A 35 °F GARAGE WITH BOTH HEATER AND CHARGER CONNECTED.
371 AMP. – SEC.
BATTERY VOLTAGE BEFORE PRESSING STARTER 6.8 VOLTS.
DURING STARTING PERIOD, VOLTAGE AVAILABLE FOR IGNITION 5.2 VOLTS
SECONDS

Fig. 14 – Starting Energy Required With and Without Motor Heater

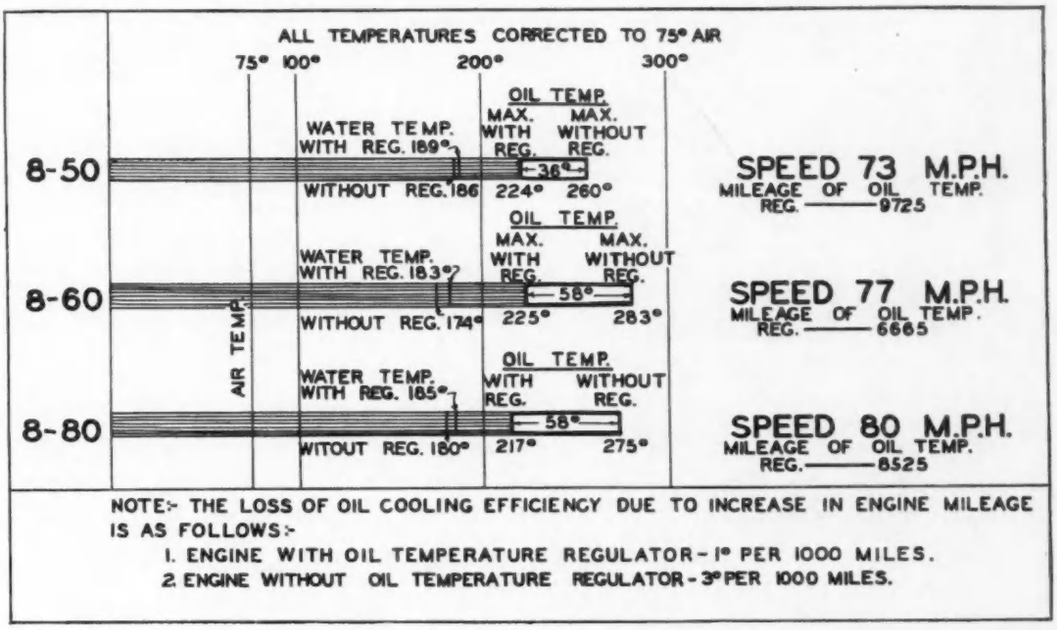

ALL TEMPERATURES CORRECTED TO 75° AIR
75° 100° 200° 300°
AIR TEMP.
8-50
WATER TEMP. WITH REG. 189°
WITHOUT REG. 186
OIL TEMP. MAX. WITH REG. MAX. WITHOUT REG.
36°
224° 260°
SPEED 73 M.P.H.
MILEAGE OF OIL TEMP. REG. 9725
8-60
WATER TEMP. WITH REG. 183°
WITHOUT REG. 174°
OIL TEMP. MAX. WITH REG. MAX. WITHOUT REG.
58°
225° 283°
SPEED 77 M.P.H.
MILEAGE OF OIL TEMP. REG. 6665
8-80
WATER TEMP. WITH REG. 185°
WITOUT REG. 180°
OIL TEMP. WITH REG. WITHOUT REG.
58°
217° 275°
SPEED 80 M.P.H.
MILEAGE OF OIL TEMP. REG. 8525
NOTE:- THE LOSS OF OIL COOLING EFFICIENCY DUE TO INCREASE IN ENGINE MILEAGE IS AS FOLLOWS:-
1. ENGINE WITH OIL TEMPERATURE REGULATOR - 1° PER 1000 MILES.
2. ENGINE WITHOUT OIL TEMPERATURE REGULATOR - 3° PER 1000 MILES.

Fig. 15 – Performance of Oil Temperature Regulator at Top Speeds

Somerville. For heating the cooling water the element is installable in the water hose and is of 100-watt capacity at 110 volts. The unit is shown in Fig. 13.

Tests of the device as reported by Mr. Somerville are shown in Fig. 14, first showing the effect of temperature on the engine without the device and, second, with it operating. These figures show that the installation in the engine is a simple matter and need not interfere with the water circulation. They also show that:

(1) When stopping the motor in a garage where the temperature is about 40 deg. fahr., the temperature of the engine block cooled from 125 deg. fahr. down to garage temperature in about 8 hr. (without heating element working).

(2) 5 min. after stopping the engine, it took 351 amp-sec. to start it again.

(3) It took 2010 amp-sec. to start the cold engine.

(4) With the heating element working, the engine-block temperature stayed above 70 deg. fahr., even after 11 hr. in a garage the temperature of which was between 30 deg. fahr. and 40 deg. fahr.

(5) It took only 371 amp-sec. to start the engine after being kept warm over night by the water heating element.

It appears from a study of this unit that, if it were doubled in capacity and equipped with an automatic thermostat, it would be generally useful and foolproof. Then, also, a similar unit could be installed readily in the sump of the crankcase to control the temperature of the engine oil.

One further detail would complete the installation. That would be a neat automatic wind-up extension cord, mounted permanently on the vehicle, for plugging in to any available current supply.

The practical use of similar heating elements has been reported to be most satisfactory by automobile owners.

An interesting means for aiding cold starting is referred to in the paper on "Lubrication and Cooling Problems of Aircraft Engines" presented by Weldon Worth before the National Aeronautic Meeting of the Society, Washington, D. C., March 12, 1937, and published in S.A.E. Transactions, July, 1937, pp. 315-324. This paper suggests the possibility of temporarily diluting the crankcase oil with gasoline at starting.

The scheme has the definite advantage that gasoline for diluting is actually available at all times, whereas electric current required for the heating element would not be available under all conditions.

Special hoods and fuel external heaters have been and are being used to some extent in severe climate to aid in starting.

Heated garages, the most commonly used means of making for easy starting, are somewhat expensive when provided specifically for that purpose.

Automatic Oil Heating and Cooling Means

Next to the preparation of the oil and engine for easy starting is the problem of elevating the oil to running temperature as quickly as possible.

Now that thermostatic control and preliminary bypass circulation of cooling water is worked out very well, it seems only logical that the system be made use of for increasing the oil temperature.

The most recent effort along this line is the extending of cylinder jackets in a number of engines to the full length of travel of the piston. Obviously this method has the virtue of being simple, effective, and low in extra cost. It does seem, however, to be lacking in capacity, having apparently about one-half the capacity actually needed.

Another effective change in design along this line was the circulation of water through the rocker-arm shaft of a valve-in-head engine. This method appears to be effective enough (including its influence upon the expansion of valve push rods) so that the builder felt justified in discontinuing the use of a special unit water-cooled oil cooler.

Air-cooled oil coolers appear to be definitely out of consideration because of their inability to perform the necessary function of warming up the oil from starting temperature.

Water-cooled oil coolers of liberal capacity, using cooling water circulated in the by-pass during warming up period, and taking the full circulation of the cooling water under heavy driving, appear to hold out a hope for the solution of the problem. Coolers of this type have been on the market for some time and have been applied to various makes of engines.

An interesting description of water-cooled oil coolers is given in a paper in the S.A.E. Transactions, April, 1931, pp. 429-437; "Oil Cooling and Oil Coolers," by W. R. Ramsaur.

Also, L. P. Saunders presents some interesting data on

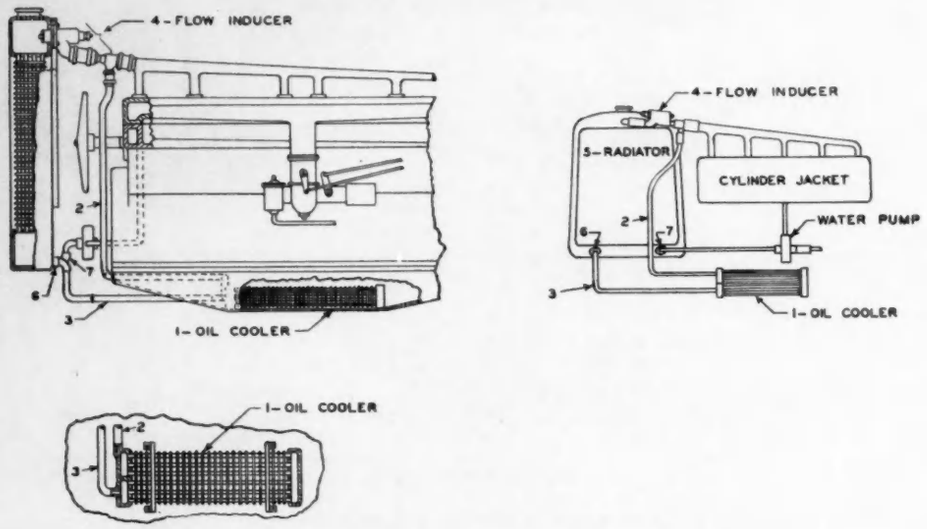


Fig. 16 – Multiflow Cooling System

similar coolers in the S.A.E. TRANSACTIONS, December, 1936, pp. 496-516; "Radiator Development and Car Cooling."

The results of a test of such a cooler applied to an automobile engine are shown in Fig. 15.

In some installations the coolers of the type and size referred to previously have apparently been lacking in cooling capacity, especially after long use. In some cases they also have been prone to leak water into the oil.

Hild Water and Oil Temperature Control

F. W. Hild, a Los Angeles engineer-inventor, was annoyed frequently by his engine overheating when ascending the toll road to the summit of Mount Wilson. Having some time on his hands, he decided to do something about it. His view of the problem was that, since the engine is now cooled externally, why not endeavor to cool it internally (namely by the oil).

His conception of means for accomplishing his purpose is shown in diagrammatic form in Figs. 16 and 17.

This system consists of a small radiator of the fin-and-tube type installed in the sump of the crankcase, the oil returning to the sump coming in contact with the external surface of the radiator. A separate water circuit is provided, the water flow being induced through the "oil radiator" by the "flow inducer," as shown, located at (4), Fig. 16, and in detail in Fig. 17. The water comes from one side of the lower tank

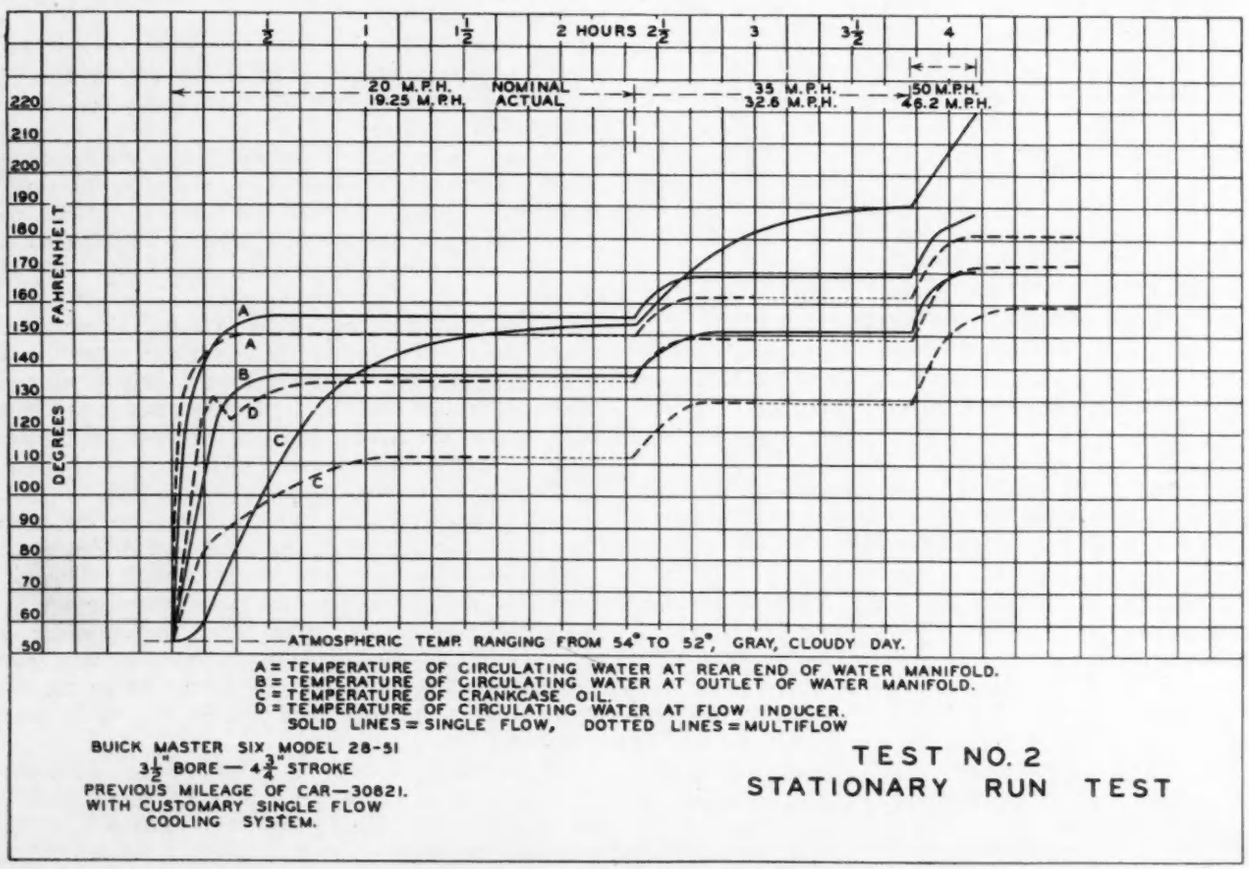


Fig. 18 – Test No. 2 – Stationary Run Test

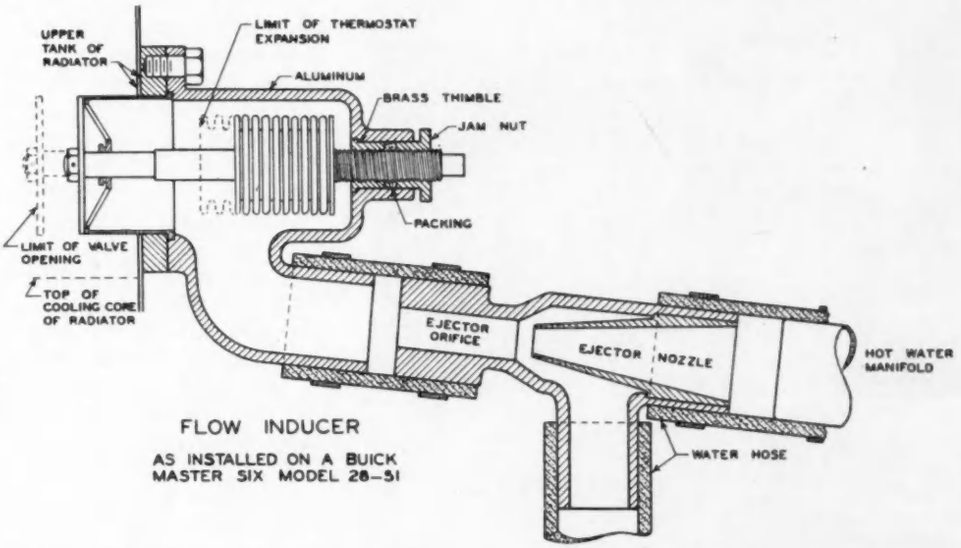


Fig. 17 – Flow Inducer – Multiflow System

of the main radiator through the "oil radiator" to the upper engine outlet circuit into the "flow inducer", and thence to the top of the main radiator with the other cooling water.

The installation described here was made on a six-cylinder automobile for trial. Tests were run with the car standing still, but with the rear wheels jacked up. Readings were taken carefully from reliable instruments.

The results of the tests are shown on Fig. 18 in which the full lines represent the car as standard or "single flow", and dotted lines represent the results of the "Multiflow" system applied to the car.

It will be seen by this data that, after 4 hr. running with standard "single flow", the crankcase-oil temperature reached 220 deg. fahr. and apparently would have continued climbing with additional running.

On the other hand, with the Multiflow system installed, the crankcase-oil temperature stabilized at 159 deg. fahr. after 4 hr. running, indicating that crankcase oil temperature was definitely under control by the Multiflow system.

Close examination of the diagram will show up some additional interesting data.

Fig. 19 shows the results of a comparative test between the

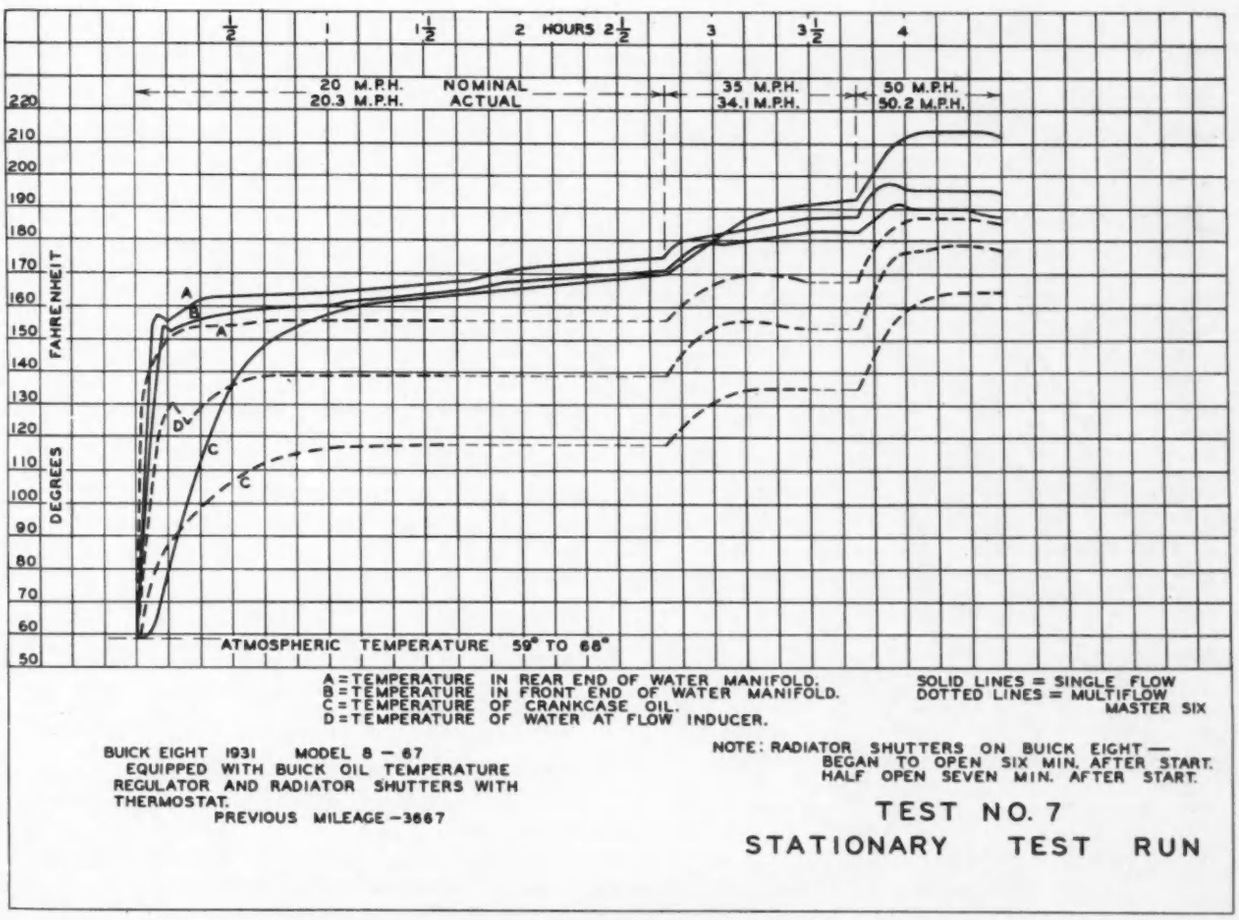

½ 1 1½ 2 HOURS 2½ 3 3½ 4
20 M.P.H. NOMINAL
20.3 M.P.H. ACTUAL
35 M.P.H.
34.1 M.P.H.
50 M.P.H.
50.2 M.P.H.
220 210 200 190 180 170 160 150 140 130 120 110 100 90 80 70 60 50
DEGREES FAHRENHEIT
A
B
A
D
C
C
ATMOSPHERIC TEMPERATURE 59° TO 66°
A = TEMPERATURE IN REAR END OF WATER MANIFOLD.
B = TEMPERATURE IN FRONT END OF WATER MANIFOLD.
C = TEMPERATURE OF CRANKCASE OIL.
D = TEMPERATURE OF WATER AT FLOW INDUCER.
SOLID LINES = SINGLE FLOW
DOTTED LINES = MULTIFLOW
MASTER SIX
BUICK EIGHT 1931 MODEL 8 – 67
EQUIPPED WITH BUICK OIL TEMPERATURE REGULATOR AND RADIATOR SHUTTERS WITH THERMOSTAT.
PREVIOUS MILEAGE – 3667
NOTE: RADIATOR SHUTTERS ON BUICK EIGHT — BEGAN TO OPEN SIX MIN. AFTER START. HALF OPEN SEVEN MIN. AFTER START.
TEST NO. 7
STATIONARY TEST RUN

Fig. 19 – Stationary Run Test – No. 7

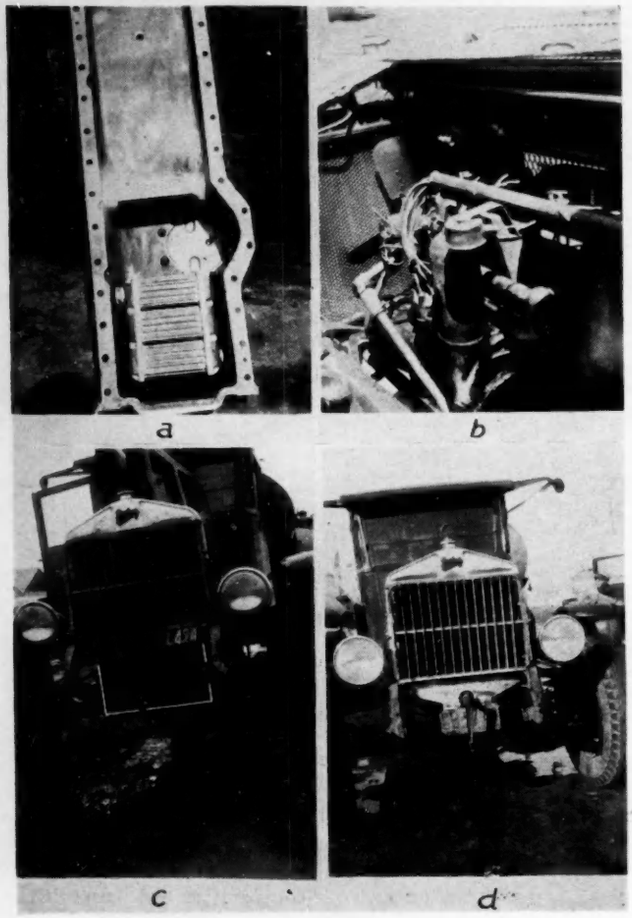


Fig. 20 (*Left*) – Multiflow System on Truck

six-cylinder automobile just referred to, equipped with the Multiflow system, and a 1931 eight-cylinder automobile of the same make equipped with a standard water-cooled oil cooler of the type referred to in Mr. Ramsaur's paper. Solid lines represent the six-cylinder car; dotted lines represent the eight-cylinder car.

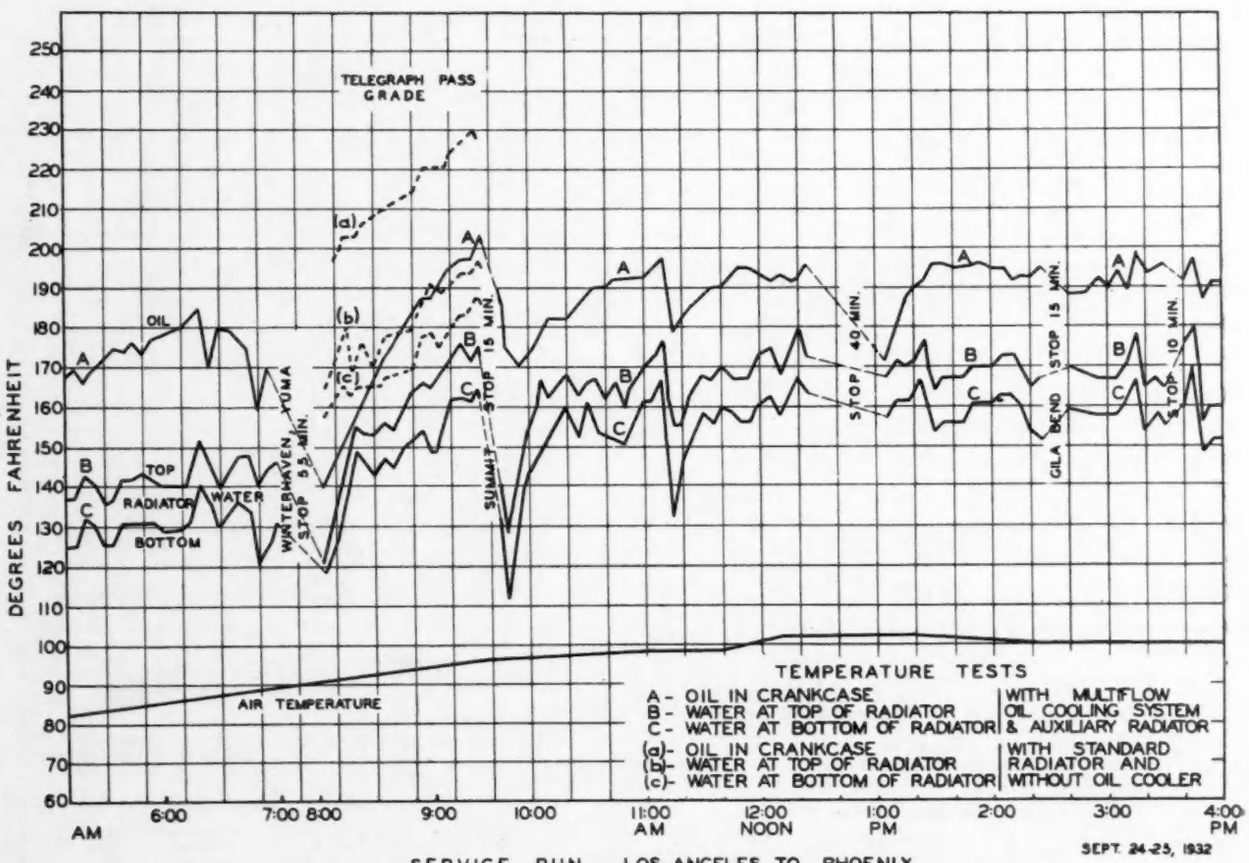
The results show that the six-cylinder engine crankcase oil stabilized at 165 deg. fahr. after four hours of running, whereas the eight-cylinder engine oil stabilized at 214 deg. fahr. in the same time. (Note the quick warm-up of the crankcase oil at the start.)

Examination of the two cooling elements involved showed that the one used in the six-cylinder car (Multiflow system) had a radiating surface of about 1532 sq. in., whereas the one in the eight-cylinder car had a surface of approximately 400 sq. in. This comparison may explain most of the difference in crankcase temperature.

Trials of the six-cylinder car on Mt. Wilson toll road with the Multiflow system were most satisfactory.

A Multiflow system was installed in a 10-ton truck (tanker) hauling nominal 68,000-lb. gross loads of gasoline from Los Angeles, Calif., to Phoenix, Ariz. Fig. 20 shows something of the installation.

Fig. 21 (*Below*) – Service Run Test of Multiflow System



SERVICE RUN LOS ANGELES TO PHOENIX
CANTLAY & TANZOLA TANKER & TRAILER - TOTAL GROSS WT 68000#
MODEL DC 27 STERLING TRUCK
6 CYL 4½" BORE, 5¾" STROKE, WAUKESHA TYPE AB
EQUIPPED WITH OIL COOLING SYSTEM

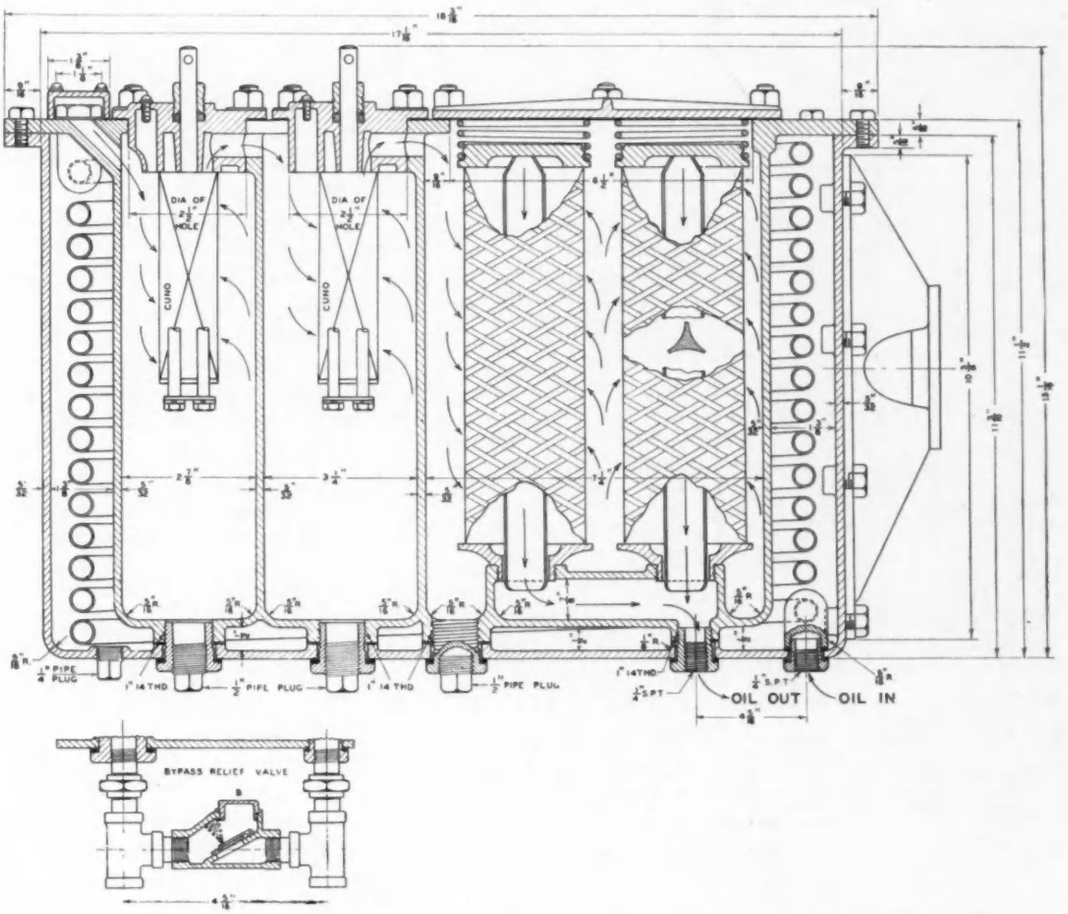


Fig. 22 – Combination Oil Filter and Temperature Regulator

The main difference between this installation and that on the six-cylinder car referred to above was the addition to the water radiator as shown at *c*, Fig. 20. This radiator was considered necessary to handle the additional heat dissipation involved in cooling the oil.

[Note: Tests by Robert W. Beal (reported in a paper presented before the Oregon Section of the Society, April, 1936, entitled "Crankcase Oil Temperature Control") indicate that about 3½ per cent of the total heat units supplied to the engine (in fuel) can be dissipated in an oil cooler.]

Trial runs were made with this 10-ton truck in its regular service. The results are shown on Fig. 21.

The peak of oil temperatures occurred at the summit of Telegraph Pass Grade, where the crankcase oil reached 230 deg. fahr. without the oil cooler, and only 203 deg. fahr. with the oil cooler.

Oil Filter and Temperature Regulator

Another unit that has come to our attention is the one designed by an engineering and sales company of Los Angeles.

The unit itself is shown in section in Fig. 22. It is a combination of a full-flow filter and an oil-temperature regulator.

The filter system consists of two disc-type automatic cleaning filters in tandem and two "polishing" filters in parallel, all enclosed in the inner case, with provision for removal and cleaning.

The temperature-regulator radiating surface consists of the external face of the inner case and a single coil of 50 ft. of ½ in. O.D. copper tubing surrounded by a water jacket. The engine cooling water circulates through this jacket, going in one side, passing all the way around, and out the other side. The radiating surface amounts to 1375 sq. in.

It is the designer's intent to place the regulator in the discharge line of the water pump. If the car does not already have a thermostat and bypass, it is his intent to install them as indicated diagrammatically in Fig. 23.

The features of this crankcase-oil temperature regulator are claimed as follows:

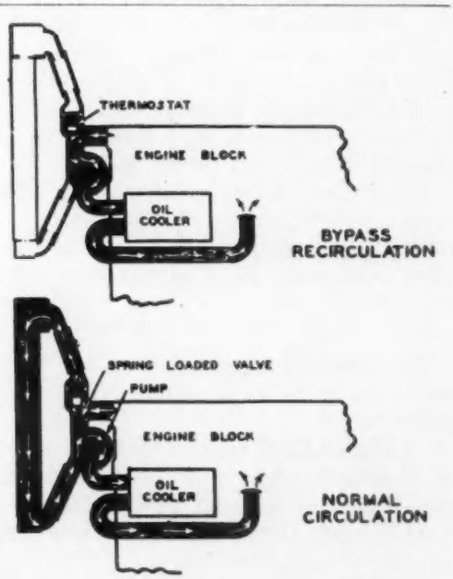
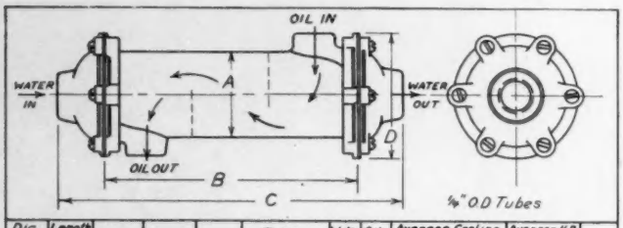


Fig. 23 – Bypass and Normal Water Circulation



Dia. of Body A	Length of Tube B	Overall Length C	Dia. Flange D	Appr. Net Wt.	Tube Surface Sq. In.	Tube Surface Sq. Ft.	Water Passage	Oil Passage	Average Cooling Capacity B.T.U. Per Min. 1° F Temp. D.	50° F Temp. D.	75° F Temp. D.	Average H.P. Rating @ 8 B.T.U./H.P. Indust. Auto.	Marine Eng.
	3"	5⅛		2.1	72	.50			.75	37.5	56.2	12.5	19
	4	6⅛		2.4	100	.69			1.03	51.7	77.5	17.2	26
2½"	5	7⅛	3⅛"	2.7	127	.88			1.32	66.0	99.0	22.0	33
	6	8⅛		3.1	155	1.08			1.62	81.0	121.5	27	41
	7	9⅛		3.4	182	1.27			1.91	95.2	142.8	32	48
	8	10⅛		3.7	220	1.45			2.18	108.7	163.0	36	54
35 Tubes							126 sq. in. = 0.0875 sq. ft.	181 sq. in. = 0.1259 sq. ft.					
	4	6⅜		4.1	230	1.60			2.40	120	188	40	62
	5	7⅜		4.7	294	2.04			3.06	153	230	51	76
3½"	6	8⅜	4¼"	5.3	358	2.48			3.72	186	279	62	93
	7	9⅜		5.9	422	2.92			4.38	219	329	73	110
	8	10⅜		6.5	485	3.37			5.05	252	379	84	126
	9	11⅜		7.1	548	3.81			5.71	286	429	95	143
	10	12⅜		7.7	612	4.25			6.38	319	479	106	160
81 Tubes							295 sq. in. = 0.2062 sq. ft.	396 sq. in. = 0.2750 sq. ft.					
	5	7⅞		8	538	3.74			5.61	281	421	93	140
	6	8⅞		9	654	4.54			6.81	341	511	114	170
	7	9⅞		10	770	5.35			8.03	401	602	133	201
	8	10⅞		11	887	6.15			9.23	461	692	154	231
	9	11⅞		12	1003	6.96			10.43	522	783	174	261
4½"	10	12⅞	5⅜"	13	1120	7.77			11.65	582	873	194	291
	12	14⅞		15	1352	9.40			14.10	705	1058	235	352
	14	16⅞		17	1585	11.00			16.50	825	1238	275	412
	16	18⅞		19	1818	12.62			18.92	946	1420	315	473
	18	20⅞		21	2050	14.22			21.35	1067	1600	355	533
	20	22⅞		23	2280	15.85			23.80	1190	1785	396	595
198 Tubes							532.5 sq. in. = 0.37 sq. ft.	607 sq. in. = 0.4219 sq. ft.					

Note: Aver. cooling capacity based on 1.5 B.T.U./Min. per sq. ft. cooling surface. Aver. H.P. rating based on 1.5 B.T.U./Min. per sq. ft. cooling (tube) surface. 50° T.D. Betw. oil & water for indust. engines—75° T.D. for marine engines, and 3 B.T.U. per min. per H.P. cooling load for both types of engines.

Fig. 24 – Type "A" Oil Cooler (Schedule)

(1) High velocity of oil through the coil, aiding in heat transfer and scouring of the inner surface.

(2) Minimum tendency for insulating film to form on the heat-transfer surface.

(3) Minimum possibility of water leaking into the oil.

(4) Economy of space by combining filter and regulator in one unit.

(5) In case of too high resistance to flow in the coil, oil may be delivered direct to the engine through the spring loaded bypass valve shown at *B* (Fig. 22, *lower left.*)

Figs. 24 and 25 show a schedule and various forms of oil coolers available on the market. Data are given showing the sizes of engines for which they are suitable.

A final suggestion worthy of consideration is that all heavy-duty units be equipped with crankcase-oil temperature gages on the dashboard. These gages should be accurate and have designated on them dangerous temperature values for corresponding S.A.E. viscosity oils. The driver can then prevent trouble by not allowing the oil temperature to exceed safe limits.

This gage is considered of even more importance than the water-temperature indicator, although the latter is important.

Conclusions

The foregoing presentation of data and factors related to the subject of crankcase-oil temperature control is summed up as follows:

(1) Automotive engine operating and laboratory experiences point definitely to the need of crankcase-oil temperature control for both low and high operating temperatures. Such control is necessary to prevent excessive wear and to improve economy and performance.

(2) Complete crankcase-oil temperature control would effect definite economies in operation, especially by permitting lighter oils to be used safely. This lubrication would allow satisfactory starting and reduce low-temperature cylinder corrosion.

(3) The lower maximum temperatures of circulating oil would have a definite influence in lowering the internal temperatures of the engine, especially those of the bearings and pistons. The life of these and related parts would thereby be prolonged.

(4) By operating the lighter oils at a controlled maximum temperature, film rupture and oil decomposition would be avoided. This operation would result in materially reduced engine wear.

(5) Many creditable efforts have been made to provide crankcase-oil temperature-control units. However, a complete mechanism for control of both high and low temperatures should be designed especially for, and built into, the engine so that the full benefit of oil-temperature-control possibilities may be realized.

(6) Until engine designers provide adequate crankcase-oil temperature control, operators may well protect their engines against excessive temperature by the use of crankcase-oil temperature indicators.

(7) Data presented here show that, by the use of engines provided with adequate oil-temperature control, S.A.E. 20 or 30 oil could be used almost universally year in and year out with a reduction in engine wear and an improvement in engine performance.

Acknowledgments

Many persons have assisted or furnished data for the preparation of this paper, including the writer's associates in the General Plant Division of the Los Angeles Department of Water and Power, and also Dr. W. F. Hamilton, H. A. Shawk, D. G. Taylor, F. M. Young, S. C. Schwarz, C. A. Chayne, F. W. Hild, and L. J. Grunder.

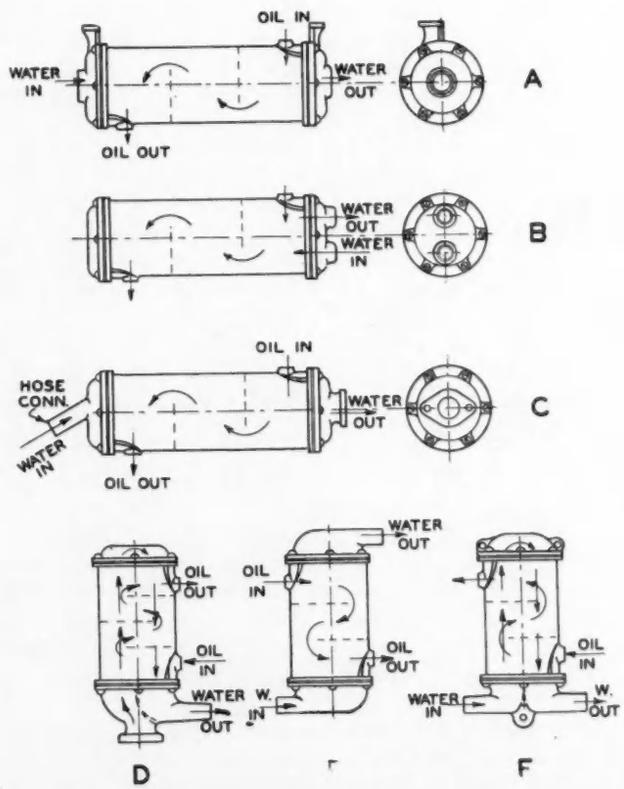


Fig. 25 – Type "A" Oil Cooler (Various Forms)

Behavior of High- and Low-Cetane Diesel Fuels

By G. C. Wilson and R. A. Rose
University of Wisconsin

THIS paper is a sequel of the paper, "Photo-Electric Combustion Analysis," presented at the 1936 Semi-Annual Meeting of the Society. The indicator described in that paper has been used to study combustion of 28 fuels and chemicals. A complete table of information of the materials used as fuels is included.

The results obtained from over 1000 oscillograms show a different shape of ignition-lag curve versus injection advance angle than it is ordinarily thought to have.

Even though the cetane values for these 28 fuels varied from 24 to 100, they all had nearly the same ignition lag when injected near the dead-center position. This minimum value is shown to be about 1/1000 sec. The fuels of higher-cetane value reach this minimum at an earlier injection angle than do those of low-cetane value.

The paper shows how a high-cetane fuel can be just as rough as a low-cetane fuel if the injection timing is too early. Moreover, a low-cetane fuel can give smooth operation if injected late enough during the compression stroke of an engine with a high compression ratio.

A COMPREHENSIVE study of Diesel fuels having ignition ratings extending over a wide range of cetane numbers requires an indicating mechanism which is extremely accurate at high speeds. In addition, the indicator must be capable of producing a large number of records to compensate for variations in successive cycles of engine operation. The apparatus developed to meet these requirements comprises a photo-electric pickup system connected to cathode-ray oscillograph tubes, with the waves recorded on photographic film revolving at a speed of 60 or more ft. per sec. A detailed description of this indicator was presented at the S.A.E. Semi-Annual Meeting in 1936.[1]

A few changes were made in some of the parts of the indicating set-up to overcome minor difficulties. In order to place the pressure-pickup piston right next to the combustion-chamber, its cylinder was bored directly in the engine head and a Midgley-type piston with a very stiff spring was installed to actuate the light shutter. Since the light shutters for both the pressure variation and the nozzle-valve stem are necessarily integral with the engine unit, vibration of the engine caused the light beam to dance over the sensitive surface of the photo-cell. The General Electric Type F J 114 was found to be unsuitable for this kind of service. An attempt was made to solve the difficulty by mounting the photo-cell rigidly on the shutter housing, but the vibration soon destroyed the photo-cell. Finally it was found that Visitron photo-cells, Type 79-A, were not sensitive to movements of the light beam and could be used successfully when mounted separately from the engine.

Test Conditions

The 3⅜ in. by 4½ in. laboratory test engine was modified to the extent of removing the plate from the top of the piston to compensate for the volume change resulting from placing the indicator piston adjacent to the combustion-chamber. During the tests covered by this paper, the compression ratio was 15.64:1. A 30-deg. pintle-type nozzle injected the fuel into the open combustion-chamber formed in the piston. The nozzle valve was kept adjusted to lift at 3100 lb. per sq. in. pressure for all fuels. The jacket water was regulated to maintain a constant outlet temperature of 140 deg. fahr. Load and speed were held as nearly constant as possible—4 hp. at 1250 r.p.m. The engine governor which controls fuel injection cut-off was in operation for all tests.

Test Fuels

The fuel samples, the characteristics of which are listed in Table 1, represent a wide selection of kinds and types. They come from all sections of the country and differ as to the source of the crude and the process of refining. Oscillograms were taken for each fuel on the list except the ones having cetane numbers lower than 24. The engine would not operate on Fuels Nos. 17, 18, 21, and 26.

The cetene numbers listed in the table were furnished by the companies supplying the fuel samples, as the samples were obtained before the standard was changed to cetane number. The cetane ratings for all samples were made by both the Delay and the C.C.R. methods in the same rating laboratory. Aniline points, gravity, and viscosity determinations were all made in the University of Wisconsin Test Laboratory. The viscosity values were determined with modified Ostwald viscometers. The spontaneous ignition temperatures (S.I.T.) were determined in a commercial testing laboratory according to the A.S.T.M. standard. It is commonly conceded that this

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 8, 1937.]

[1] See S.A.E. TRANSACTIONS, November, 1936, pp. 459-468; "Photo-Electric Combustion Analysis," by R. A. Rose, G. C. Wilson, and R. R. Benedict.

Table 1 – Test Fuel Data

FUEL NO.	NAME	A.P.I.	IGNITION RATING CETENE	CETANE DELAY	CETANE C.C.R.	ANILINE POINT	DIESEL INDEX	S.I.T. °F	DISTILLATION TEMP F INIT. POINT	10%	50%	90%	END POINT	% OFF 600 °F	VIS. 100F S.U. SEC.	% RES. A.S.T.M.	DESCRIPTION	CRUDE	PROCESS	OBTAINED FROM
1	NO.1 FURNACE OIL	41.0	60	50	60	147	60.2	426	355	400	449	510	515	100	32.7	0	GAS OIL	UNKNOWN	STR R. DISTIL	CONKLIN & SONS LOCAL MARKET
2	SPL. AUTO. DIESEL	38.7	70	62	73	186	73.2	412	500	536	561	612	660	85	41.2	0	1933 DIESEL FUEL			STANDARD OIL OF NEW JERSEY NEW YORK
3	ESSO DIESEL 208	38.4	70	53	68	160	81.4	426	338	430	495	590	637	92	34.7	0.6	1936 DIESEL FUEL			
4	LOW REF. FUEL	14.8	22-25	19	24	56	83	748	386	523	588	690	723	57	46.7	8.8	GAS OIL	GULF COAST	CRACKED DISTIL	TEXAS CO. NEW YORK
5	DIESEL FUEL 335	35.2	55	39	52	120	43.4	457	402	428	445	472	516	100	32.2	0.2	DIESEL FUEL	MID CONT.	" "	MID. CONT. PET. CORP. TULSA, OKLA.
6	DIESEL FUEL 46	36.1	63	45	55	142	51.3	439	324	418	482	568	640	94	34.2	2.2	DIESEL OR GAS OIL	"	CRACKED DIS.RS.	
7	DIESEL FUEL	32.6	70	47	64	172	56.1	435	346	340	616	668	690	40	36.6	0.5	DIAMOND	"	STR. RUN DISTIL.	
8	HIGH REF. FUEL	43.7	78	70	87	173	75.5	424	378	424	496	595	676	90	34.5	1.0	A- GAS OIL	SO. ILL.	STR R. DISTIL.	SHELL PET. CORP. WOOD RIVER, ILL.
9	GAS OIL	37.4	59	56	73	170	63.5	432	400	480	540	640	702	79	38.0	0.5	B- "	MID. CONT.	"	
10	DIESEL FUEL	41.7	84	51	62	161	66.9	469	415	484	565	660	711	72	40.0	0.7	C- "	MID. CONT. & HENDRICKS	"	
11	NO.4 BURNER OIL	26.2	40	39	44	130	36.6	504	384	447	528	—	—	82	38.6	5.6	D- BOTTOMS	"	CRACKED	
12	NO.3 BURNER & M.NAPH.	27.5	30	31	34	82	22.5	572	400	431	457	572	596	100	32.2	0.3	E- DISTILLATE	MIXED RES. & M.NAPTH	"	
13	DIESEL FUEL 355	36.8	50+	52	73	174	64.2	415	378	482	549	624	668	84	38.1	0.2	GAS OIL	MID. CONT.	ST. RUN	SINCLAIR REF. CO. EAST CHICAGO
14	DIESEL FUEL 160	42.0	60+	44	69	156	66.6	424	340	396	450	550	615	97	32.6	0.2	CRUDE KEROSENE	"	"	
15	DIESEL FUEL	31.1	48	44	49	130	40.4	455	324	458	502	592	688	91	35.6	1.1	GAS OIL	PENN.	CRACKED	
16	DIESEL FUEL	38.8	62	56	78	181	69.8	419	381	324	582	640	677	62	40.0	0.3	"	"	STR. RUN	
17	DISTILLATE	14.2	20	TOO LOW	18	—	—	796	398	492	534	638	708	82	38.6	2.5	DISTILLATE	SAN JOAQUIN VALLEY	CRACKED	UNION OIL CO. OF CAL. WILMINGTON, CALIF.
18	EXTRACT BOTTOMS	23.0	30	TOO LOW	16	18	4.14	750	426	440	457	501	558	100	33.0	0.8	BY PRODUCT FROM KEROSENE	CALIF.	EDELEANU TREATED	
19	GAS OIL	25.0	45	29	36	138	34.9	561	318	482	600	742	760	60	53.5	2.0	GAS OIL	SAN JOAQUIN VALLEY	STR.R.DISTIL.	
20	GAS OIL	38.4	56	48	59	158	52.8	439	446	490	553	678	760	72	40.7	1.1	" "	LOS ANGELES BASIN	" " "	
21	160 - OCTANE	70.6	17	—	23	192	127.8	804	207	209	210	212	235	98.3	29.2	0.1	PURE CHEMICAL			ROHM & HAAS CO. PHILA, PA.
22	CETENE	47.8	100	67	82	144	68.4	424	488	510	530	540	550	98.1	35.5	0	" "			E.I.DUPONT, DE NEMOURS & CO. WILMINGTON, DELAWARE
23	CETANE	51.0	—	100	100	200	108.0	392	405	540	545	545	545	90.9	35.9	—	" "			
24	N. HEPTANE	52.5	—	57	57	156	81.9	430	208	208	215	250	262	95.0	29.5	—	CHEMICAL	JEFFREY PINE		CAL. TURP PRODUCTS, NEVADA CITY, C
25	KEROSENE	42.5	51	53	71	160	68.0	426	336	405	448	490	522	98.7	32.6	0.1	PERFECTION	MID-CONT	STR. RUN	STAND. OIL OF IND., WHITING, IND.
26	ALPHA M. NAPH.	7.5	—	0	0	-5	—	997	457	480	486	470	474	99.0	32.7	—	PURE CHEMICAL			REILLY TAR & CHEMICAL CO.
27	MINERAL OIL LT.	39.5	—	70	94	215	85.2	—	521	600	675	720	735	10.0	53.5	0.1	U.S.P. MEDICINAL	PENN		FILMITE CORP. MILW., WIS.
28	GASOLINE - REF. A	61.2	—	35	54	130	79.5	—	160	220	255	310	374	99.0	29.4	0	FOR OCTANE RAT.	PENN	STR. RUN	

standard is not satisfactory for testing Diesel fuels because of the departure from conditions of operation, the test being performed by dropping the fuel upon a hot plate surrounded by air at atmospheric pressure, but the values are included here because of the lack of a better standard. The per cent residue was determined by the Waukesha method of open-cup combustion.

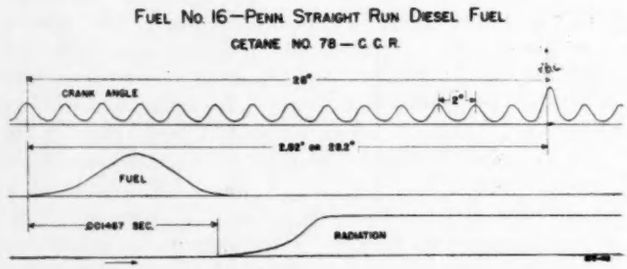


Fig. 1 – Oscillogram Showing Constancy of Speed During One Engine Revolution

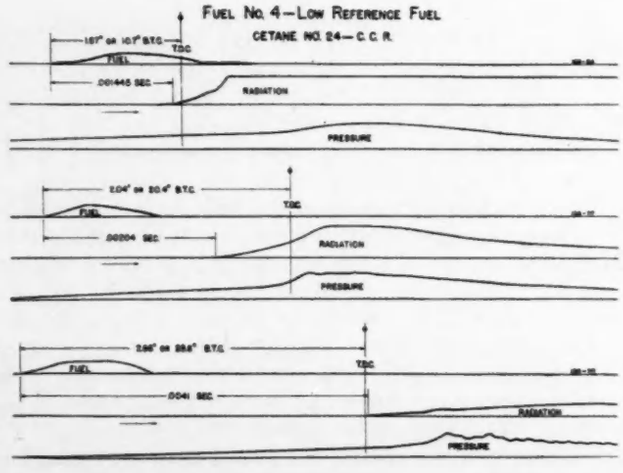


Fig. 2 – Oscillograms for Low-Cetane Fuel

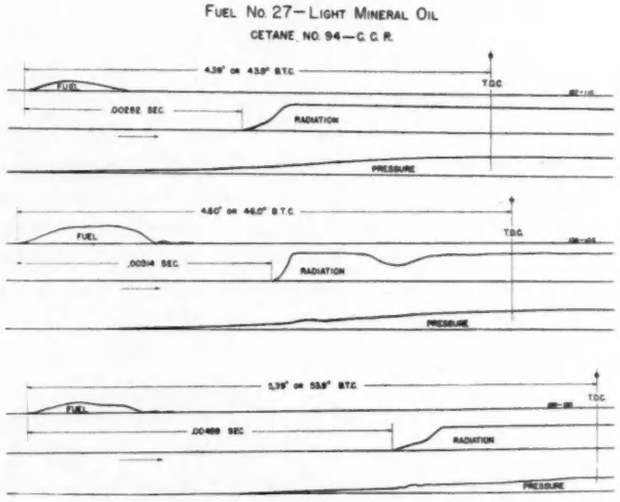


Fig. 3 – Oscillograms for High-Cetane Fuel

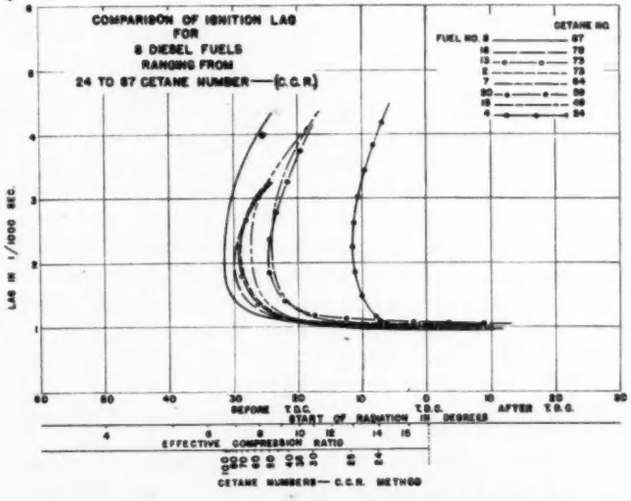

COMPARISON OF IGNITION LAG FOR 8 DIESEL FUELS RANGING FROM 24 TO 87 CETANE NUMBER—(C.C.R.)
CETANE NO.
LAG IN 1/1000 SEC.
BEFORE T.D.C.
START OF RADIATION IN DEGREES
AFTER T.D.C.
EFFECTIVE COMPRESSION RATIO
CETANE NUMBERS— C.C.R. METHOD

Fig. 4 – Effect of Fuel Ignition Quality upon the Beginning of Combustion

Many of the samples tested were not intended for use as Diesel fuels, some of them being pure chemicals; nevertheless, the differences in these materials furnish the basis for an explanation of the behavior of fuels in the combustion-chamber.

Discussion of Results

There was some uncertainty about the accuracy of measuring the angle of injection advance from the film which was driven at a constant speed in synchronism with the engine. The reason for this uncertainty was based upon the fact that the rotative speed of the engine must slow down during the compression stroke and speed up during the power stroke. A check on the accuracy of this method of measurement was obtained with the aid of a slotted disc mounted on the engine shaft. The slots were out near the rim and spaced at 2-deg. intervals, with a long slot at the dead-center position. A photo-cell was mounted on one side of the disc, and a light on the other side. The impulses were recorded by the oscillograph in place of the pressure waves. Fig. 1 shows a sample of one of these oscillograms. These records show no appreci-

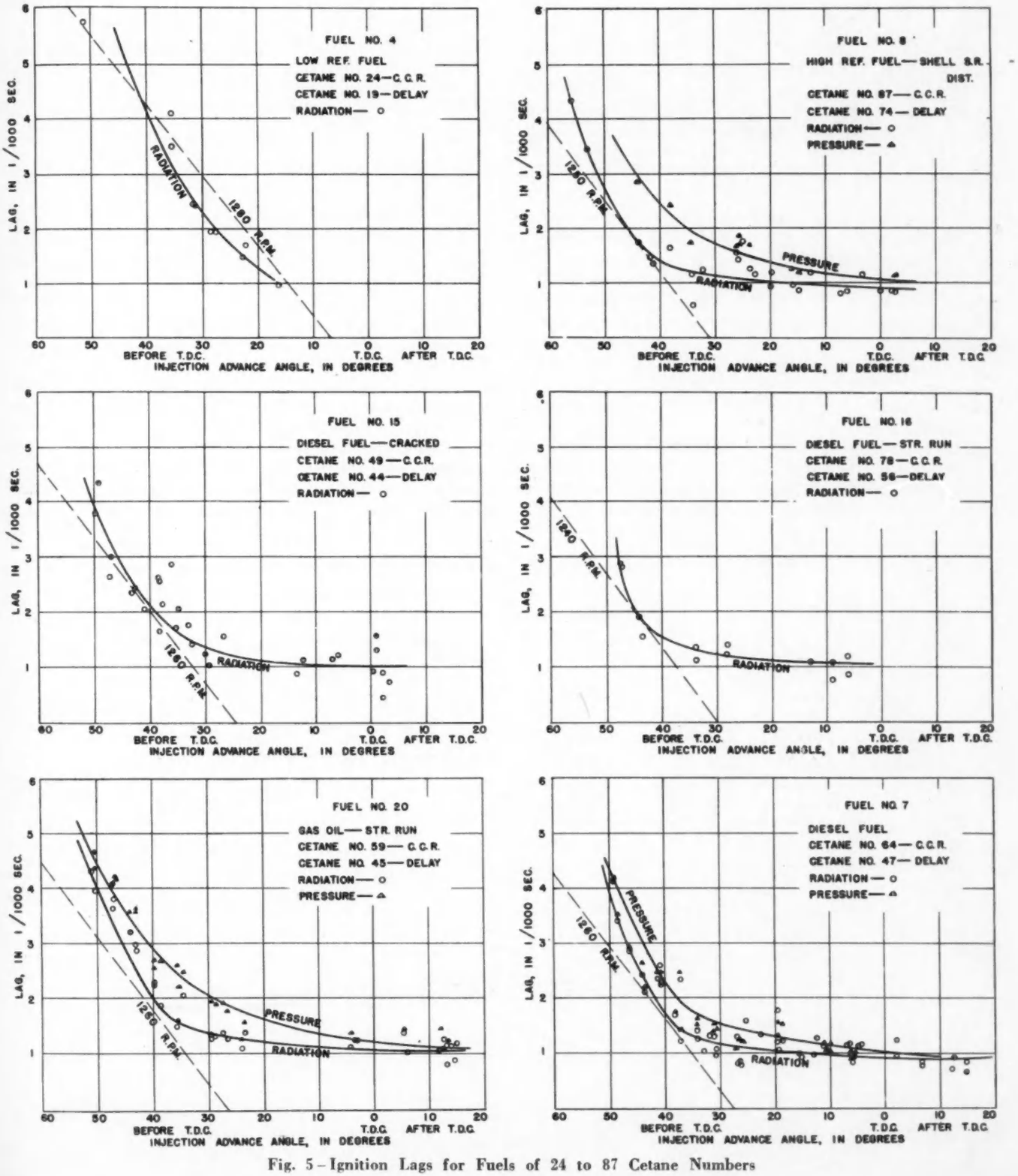


Fig. 5 – Ignition Lags for Fuels of 24 to 87 Cetane Numbers

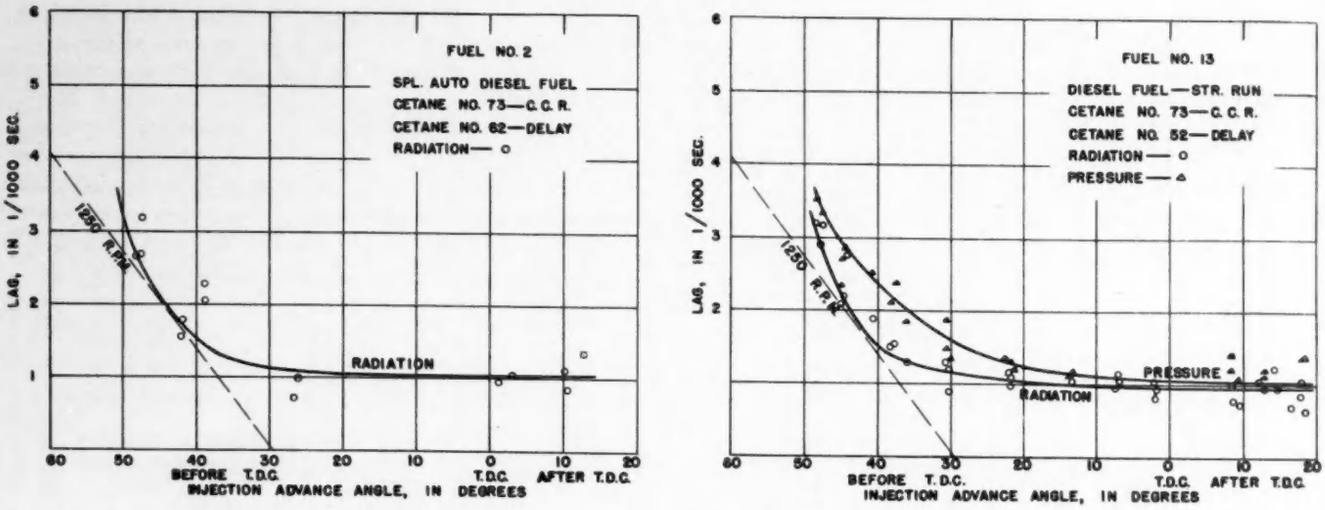


Fig. 6 – Comparison of Ignition Lags for Two Fuels of Same Cetane Number

able change in flywheel speed during the period of fuel injection-advance angle. This test also checked the accuracy of the spark-actuating mechanism, for it showed the spark dot always within 1 deg. of the high wave caused by the long slot at dead-center. A shutter driven at half-speed cut out the action of the disc slots during the exhaust and intake strokes.

Figs. 2 and 3 show sample oscillograms for fuels of low- and high-cetane ratings, respectively. Over 1000 such diagrams were taken and measured to determine accurately the time elapsing between the injection of fuel and the start of combustion, measured both with reference to radiation and to pressure rise. Some of these results have been used for plotting curves. Fig. 4 shows how combustion will start earlier in the compression stroke for fuels of high cetane rating. It also shows that, when a fuel is injected too early, the effect is to increase the ignition-lag period. The fuels line up very definitely in the order of their cetane rating. These same ignition-lag data have been plotted against angle of fuel injection and are shown in Figs. 5 and 6. These curves show the time lag for fuels ranging in values of C.C.R. cetane numbers from 24 to 87 and injected at crank angles varying from more than 50 deg. before top-center to 20 deg. after. It is apparent that successive cycles are not all exactly alike; hence, it is necessary to have a large number of points to establish the curve with reasonable accuracy. It is also apparent that some fuels show a greater difference than others between the time radiation starts and the beginning of pressure rise, whereas some fuels show no measurable difference at all between these events. The fuels for which no pressure curves are shown were

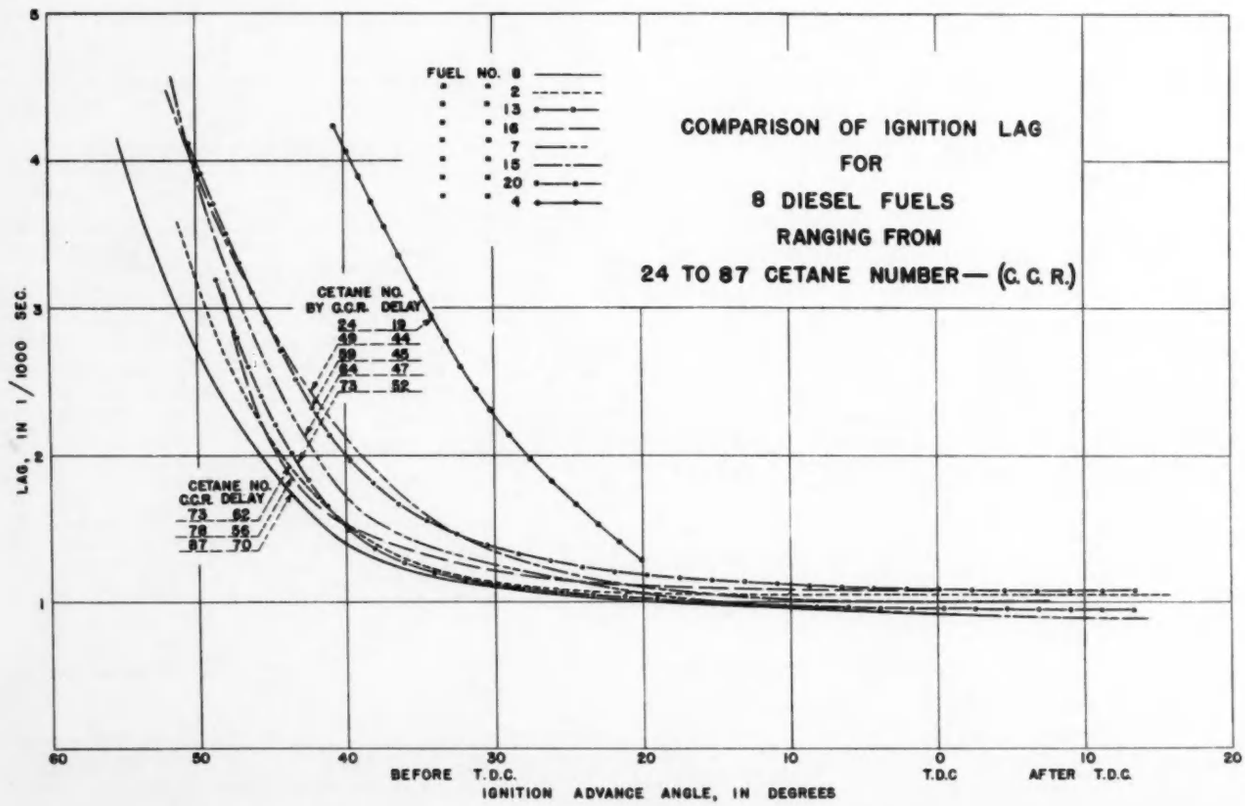


Fig. 7 – Comparison of Ignition Lags for Fuels of Different Cetane Numbers

used in the engine during the tests in which the third beam was utilized to indicate the crank position.

Fig. 7 shows a composite picture of all the radiation curves, reproduced from Figs. 5 and 6. Several significant facts are brought out by this set of curves. It will be observed that all of the curves indicate a minimum ignition lag of about 0.001 sec. The point at which they leave this minimum ignition lag is approximately in the order of their cetane rating. The fact that some of the curves cross may possibly be due to inaccuracy in locating the curves, but it seems more probable that this peculiarity is due to differences in the fuels. Such a peculiarity is indicated by the inconsistent difference between the cetane ratings by the C.C.R. and the Delay methods.

The engine used for these tests had a compression ratio high enough to burn low-cetane rating fuels, so that injection of fuel later in the compression stroke was the equivalent of injecting it into an engine of higher compression ratio. This tendency of high compression ratio to nullify the differences in ignition qualities of fuels has been observed by MacGregor.[2]

Theoretical Analysis

The shape of each curve in Fig. 7 shows a striking resemblance to two straight lines, one of which is practically coincident for all fuels due to the fact that all of them show practically the same minimum ignition lag. The point of intersection with the other straight line might be termed the break-away point. This point comes earlier in the compression stroke for the fuels of higher cetane rating. This condition leads to a theoretical consideration of reasons which might account for the shape of these curves. The chart shown in Fig. 8 has been derived from a theoretical consideration for the special test engine by combining its effective compression ratio at different crank positions with the cetane value determined by the C.C.R. method. It is surprising to find that the effective compression ratios in the test engine correspond very closely with those of the C.F.R. test engine in which the fuels were rated. It can hardly be expected that all engines would show such close correlation although, in general, there should be a definite relation. The graph at the lower right shows the relation between cetane numbers and critical compression ratio (C.C.R.) for the C.F.R. engine in which the fuels were rated. The lower-left curve was constructed by calculating the compression ratio for different positions of the test-engine piston as it approaches top dead-center. The upper graph shows ignition lag, either in time units or in degrees of crank travel at a speed of 1200 r.p.m., plotted against crank angle. Thus both axes are virtually time axes.

To use the chart of Fig. 8, take any cetane number fuel, say 30 at point *A*, and find the C.C.R. of the C.F.R. engine corresponding to this rating by tracing the path *A* to *B*. Then trace horizontally from *B* to *C* to find the crank position in the special test engine at which the effective compression ratio is the same. The crank angle is found at *D* to be 18 deg., which is the earliest point of advance that a fuel of 30 cetane number will burn with minimum delay. If the fuel is to fire at this point, it must be injected 0.001 sec. earlier to allow for the minimum ignition lag. Even under conditions of temperature and pressure well above the critical, 0.001 sec. elapses before ignition occurs. In general, when fuel is injected into air below the critical conditions, it simply waits until the proper temperature and pressure are reached before it starts its preparation for burning. Fuel injection earlier than this time increases the ignition-lag period that much more. The slope of this line from *D* is determined by the scale of the coordinates. If both scales were the same, the slope of the line would be 45 deg. because the lag before ignition and the interval for crank travel are the same period of time. Similarly, lines for fuels of other cetane ratings have been drawn. If ignition lag could be considered as only a waiting period, these diagonal lines would show the increase in ignition lag for fuels of different cetane rating. Corresponding diagonal lines have been drawn in Figs. 5 and 6 to show how closely they agree with the experimental results for the average engine speed during the time the test data were recorded.

Using an injection advance angle of 37 deg. before top-center, a 75-cetane fuel would have an ignition lag of 0.001 sec., and a 24-cetane fuel, an ignition lag of over 0.004 sec. Due to the long delay for the latter, a great deal of vapor can mix thoroughly with the air and, when ignition occurs, combustion will be very rough. Spontaneous ignition probably starts in many places in this case instead of beginning in the fuel jet and spreading gradually.

By observation during test runs it is possible to block off an approximate ignition-delay range within which the engine will operate without roughness. If the delay period is shorter than about 0.0015 sec., smooth combustion results and, if the delay exceeds about 0.0027 sec., the operation of the engine is very rough regardless of how high the cetane rating of the fuel may be. Of course, the higher the cetane rating, the earlier the fuel must be injected to cause so great a delay period. This condition is shown by the oscillograms in Figs.

[2] See S.A.E. TRANSACTIONS, June, 1936, pp. 217-224; "Diesel Fuels – Significance of Ignition Characteristics," by J. R. MacGregor.

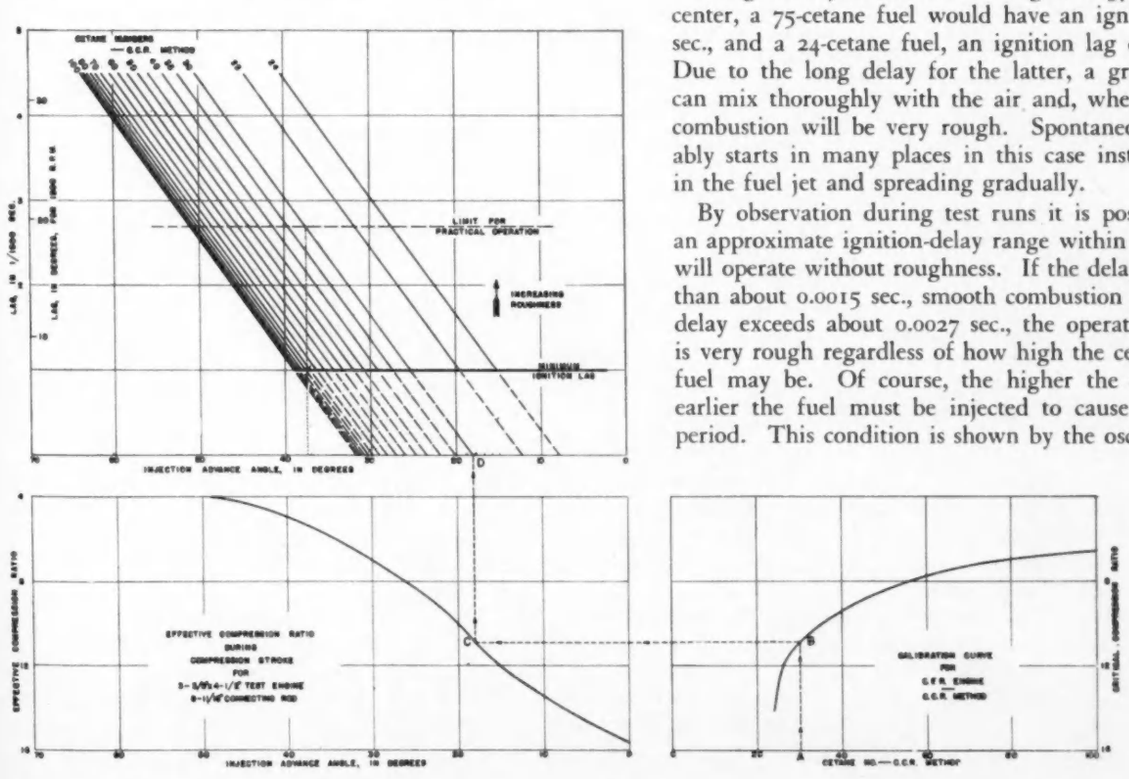

LIMIT FOR PRACTICAL OPERATION
INCREASING ROUGHNESS
MINIMUM IGNITION LAG
INJECTION ADVANCE ANGLE, IN DEGREES
EFFECTIVE COMPRESSION RATIO DURING COMPRESSION STROKE
CALIBRATION CURVE FOR C.F.R. ENGINE — C.C.R. METHOD
CETANE NO. — C.C.R. METHOD

Fig. 8 – Chart for Determining Injection Advance Angle for Smooth Diesel Operation

2 and 3, where roughness is indicated by irregularities in the pressure waves. Although these irregularities are undoubtedly the result of indicator oscillations, it is believed that the combustion was necessarily rough in order to set up these oscillations. These figures represent rough and smooth fuels, respectively. For smooth engine operation, the chart of Fig. 8 shows the earliest angle of advance for fuel injection to be 27 deg. for the 24-cetane fuel, and 53 deg. for the 94-cetane fuel as read at the intersection of the horizontal line showing "Limit for Practical Operation."

In Fig. 2 with late injection, the 24-cetane fuel burns smoothly, but as the time of injection is advanced beyond 27 deg. before top-center, combustion becomes very rough. Similarly, Fig. 3 shows that, even with the 94-cetane fuel, injection earlier than the critical angle of 53 deg. before top-center causes very rough engine operation.

It is hardly to be expected that all fuels will conform to this theoretical analysis, but some such basis for fuel classification seems to be practicable. Tests of gasoline show that the minimum ignition lag is substantially the same as for other fuels but, in contrast, no measurable difference exists between the time combustion radiation starts and the beginning of pressure rise. With regard to the time required for vaporization, apparently the higher volatility of gasoline is not a factor affecting the time required for preparation of the fuel before it ignites.

Reducing or increasing the compression ratio of the engine has the effect of shifting the entire family of curves in Fig. 8 to the right or left respectively. It only changes the ignition lag for fuel injection advance earlier than the point at which minimum ignition lag begins.

Some of the information of Table 1 has been rearranged to make Table 2. All fuels in Table 2 are arranged in the order of their cetane rating by the C.C.R. method. This arrangement permits easy observation of the coordination of the various fuel properties with ignition rating. A column has been added to this table showing minimum ignition lag periods in one-thousandths of a second.

Conclusions

The following conclusions are necessarily based on the results obtained when operating the small high-speed direct-injection test engine with the 25 fuels included in these tests. However, it is reasonable to suppose that similar combustion behavior occurs in other Diesel engines.

(1) When any fuel is injected into the combustion space of a Diesel engine, the ignition delay period is about 0.001 sec. if the air is at or above the critical ignition pressure and temperature.

(2) All fuels give rough operation if injected as much or more than 0.0027 sec. before combustion starts. Thus, a low-cetane fuel may burn smoothly, and a high-cetane fuel may burn roughly, depending upon the angle of injection advance.

(3) The outstanding cause of ignition delay and engine roughness with low-cetane fuels is too early injection of the fuel. The time of beginning of fuel injection should not only vary with the engine speed, but it should be adjustable to suit each different fuel.

Acknowledgment

This paper was made possible by the support furnished by the Wisconsin Alumni Research Foundation and the University of Wisconsin College of Engineering. Dean E. B. Fred of the Graduate School, Dean F. E. Turneaure of the Engineering College, and Prof. G. L. Larson of the Mechanical Engineering Department, have all extended full cooperation which is gratefully acknowledged. The whole-hearted assistance of various graduate and undergraduate students who have participated in this project, especially Mr. Riggert, Mr. Griffith and Mr. Jedeka, is deeply appreciated. To Prof. L. A. Wilson who revised the manuscript and all who have taken part, the authors are duly grateful.

Table 2 – Test Fuel Data Arranged in Order of C.C.R. Cetane Rating

IGNITION RATING						DISTILLATION TEMP. °F										
CETANE		Cet-	Aniline	Diesel	S.U.T.	Init.				End	% Off	Vis. 100°	%		Fuel		Min. Ign. Lag
C.C.R.	Delay	ene	Point	Index	°F	Point	10%	50%	90%	Point	600 °F	S.U. sec.	Res. Aq.Cq.	NAME	No.	A.P.I.	Sec./1000
100	100	--	200	102.0	392	405	540	545	545	545	90.9	35.9	---	Cetane	23	51.0	0.9
94	70	--	215	85.2	---	521	600	675	720	735	10.0	53.5	0.1	Mineral Oil Lt.	27	39.6	1.0
87	70	78	173	75.5	424	378	424	496	595	676	90	34.5	1.0	High Ref. Fuel	8	43.7	0.9
82	67	100	144	68.4	424	488	510	530	540	550	98.1	35.5	0	Cetene	22	47.5	1.0
78	56	62	181	69.8	419	381	324	582	640	677	62	40.0	0.3	Diesel Fuel	16	38.6	1.0
73	52	50+	174	64.2	415	378	482	549	624	668	84	38.1	0.2	Diesel Fuel 355	13	36.8	1.0
73	56	59	170	63.5	432	400	460	540	640	702	79	38.0	0.5	Gas Oil	9	37.4	1.2
73	62	70	188	73.2	412	500	536	561	612	660	85	41.2	0	Spl. Auto Diesel	2	38.7	1.0
71	53	51	160	68.0	426	335	405	448	490	522	98.7	32.6	0.1	Kerosene	25	42.5	1.3
69	44	60+	156	65.6	424	340	395	450	550	615	97	32.6	0.2	Diesel Fuel 160	14	42.0	1.0
68	53	70	160	61.4	426	338	430	495	590	637	92	34.7	0.6	Esso Diesel 208	3	38.4	1.0
64	47	70	172	56.1	435	346	540	616	668	690	40	36.6	0.5	Diamond Fuel	7	32.6	0.9
62	51	54	161	66.9	469	415	484	555	660	711	72	40.0	0.7	Diesel Fuel	10	41.7	0.9
60	50	60	147	60.2	426	355	400	449	510	515	100	32.7	0	No. 1 Furnace Oil	1	41.0	0.9
59	45	56	158	52.8	439	446	490	553	678	760	72	40.7	1.1	Gas Oil	20	33.4	1.0
57	57	--	156	81.9	430	205	208	215	250	262	95.0	29.5	---	N. Heptane	24	52.5	1.4
55	45	63	142	51.3	439	324	418	462	568	640	94.0	34.2	2.2	Diesel Fuel 46	6	36.5	1.2
54	35	--	130	79.5	---	160	220	255	310	374	99.0	29.4	0.0	Gasoline - Ref. A	28	61.2	1.0
52	39	55	120	43.4	457	402	428	445	472	516	100.0	32.2	0.2	Diesel Fuel 335	5	36.2	1.0
49	44	48	130	40.4	455	324	458	502	592	688	91.0	35.6	1.1	Diesel Fuel	15	31.1	1.0
44	39	40	130	36.6	504	354	447	528	---	---	82	38.6	5.6	No. 4 Burner Oil	11	28.2	1.0
36	29	45	139	34.9	561	318	482	600	742	760	60	53.3	2.0	Gas Oil	19	25.0	1.3
34	31	30	82	22.5	572	400	431	457	572	596	100	32.2	0.3	No. 3 Burner & M. Naph.	12	27.3	1.0
24	19	22-25	56	8.3	748	386	523	588	690	723	57.0	46.7	8.8	Low Ref. Fuel	4	14.8	1.0
23	--	17	182	127.8	804	207	209	210	212	235	99.3	29.2	0.1	Iso - Octane	21	70.6	No Test
18	Too Low	20	---	---	795	398	492	534	638	708	82.0	39.6	2.5	Distillate	17	14.2	No Test
16	Too Low	30	18	4.14	750	426	440	457	501	558	100	33.0	0.3	Ext. Bottoms	18	23.	No Test
0	0	--	-5	---	997	457	460	466	470	474	99	32.7	---	Alpha N. Naph.	26	7.5	No Test

Automobile Design and Safety

By John H. Hunt
General Motors Corp.

PRESENTING a cross-section of the constructive thought of Society members on motor-vehicle design from the standpoint of highway safety, this paper deals with progress that is being made in present-day cars and offers pertinent suggestions regarding possible improvements for the future.

DESTRUCTION of life and property in highway accidents has developed a demand for action. Only a small percentage of accidents are charged against the vehicle, and in most of the accidents in which the vehicle is alleged to have been a contributing cause, it is the maintenance of the vehicle, not the design which is reported to be at fault. The car designer cannot, however, ignore this situation.

First – the number of accidents will be reduced not by a striking change in any one factor, but by the greatest possible reduction in each cause that can be definitely determined. Second – the car designer is already being attacked because of certain characteristics of his product, and once the public realizes the opportunity, drivers in accidents will increasingly claim the accidents were caused by some design fault, and such claims will get into the statistics. Third – we have the strongest commercial motive to assist in any way possible and even if outside our recognized responsibilities. These victims are our customers. For everyone killed, an unknown number are being scared away from a full use of our product.

This paper was presented at the request of the Passenger Car Activity Committee which is very much alive to the situation, and to the need of more active participation of the car designer in the general study of safety measures. Opinions and data were solicited from the various engineering organizations in the industry, and a general response received. Anything of merit in this discussion is due to these contributions. If the discussion seems at fault, I have failed to analyze this information properly, without stating as I have done in some cases, that I am giving my own opinion on a controversial subject. It seemed advisable to divide the broad subject into certain subdivisions for convenience in discussion.

It has been a disappointment to find that so little discussion could be based upon the type of numerical analysis which the successful engineer uses in attacking a problem. We simply lack data on which to base any analysis. Certain design faults are alleged to exist, but we have no information as to whether cars which seem to have more of these faults than others are involved in more accidents per mile of service than other cars. We must try to help the safety investigators collect the data so that more accurate deductions can be made.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 6, 1937.]

Lighting

Automobile headlighting has been continuously attacked as an important source of highway danger. The same critics claim that our headlights give insufficient road illumination, and at the same time cause glare. Many of these critics seem to feel that the car engineer is at fault. It is hardly reasonable to hold the engineer responsible for the limitations of the human eye.

At the present time the regulations permit an automobile engineer to use substantially the same amount of light as that obtainable from a 60 watt lamp to illuminate the roadway in front of a car. The driver should be able to see substantial objects on the roadway at least 300 ft. ahead on a really dark night. Few would object to a requirement that a strip at least 40 ft. wide be illuminated. This gives us an area of 12,000 sq. ft. Not many of us would be satisfied with one 60 watt lamp in a room 15 ft. square or containing 225 sq. ft. Good highway lighting is believed to require 2 to 3 cp. per linear ft. Making no allowance for the fact that some light must be above the roadway at 300 ft. if an object is to be seen at that distance, and that it is technically impossible to distribute the light properly over a strip as defined, the level of illumination on the roadway must be minute compared to that we expect in our homes, and the driver is at a relatively great distance from the object to be seen. The eye must adjust itself to viewing objects at a very low level of illumination, and when so adjusted, the eye is exceedingly sensitive to the glare effect of bright sources of illumination in the field of vision. The angle between a line from the driver's eye and an object directly in front of him, and from the same eye to the headlamps of an approaching car using the same roadway, is very small.

It is impossible to provide a source of light which will illuminate the roadway at any appreciable distance ahead of the car without at the same time glaring the eyes of the driver of an approaching car.

Our headlighting troubles are very much complicated by the persisting effect of specifications developed years ago when conditions were quite different.

The lamps available were single beam lamps having plain front glass closures, using narrow concentrated beams, and in the absence of regulations were aimed in such a way as to provide the maximum visibility on a level road, which also caused intolerable glare. Speeds were relatively low, the legal limit being rarely as great as 35 m.p.h., and the roads were such that speeds in excess of this were not usually attempted

at night by the average driver. There was still a large amount of horse-drawn traffic on the highway. The car manufacturers made no effort to improve the situation. There was a great public outcry against glare and attempts were made to meet the situation by drawing up specifications limiting the amount of light which could be directed above the headlamp level. Existing lamps so aimed were found to be practically useless, and distributing lenses made their appearance.

Inspection of headlamps by state and municipal authorities began. These authorities took the position that the car must never give a glaring light when fully loaded, and applied so-called loading allowances, which required that the lights of the empty car be so aimed that the effect of the load could not raise the light beam above the horizontal. Since most cars are driven a great deal without rear seat passengers, the effect was to limit the driver to a light which was not only inadequate, but also gave the impression of diving into a black pit over a brightly lighted foreground, thus contributing to driver fatigue. The law usually required that a substantial object on the roadway 200 ft. ahead should be made visible by the headlamps. It is believed that this requirement was not met after the lamps were aimed according to the loading requirements as far as light going directly from the lamp to the object is concerned. Under very favorable pavement conditions the object would be lighted, but not under unfavorable conditions.

We cannot blame the regulating officials too much. They were directed to meet two opposing requirements under conditions where both requirements could not be met simultaneously. They elected to overlook failure in one direction in order to meet the condition for which there was the most public demand. Engineers make the same type of decisions every day.

Multiple-beam systems were developed, based on the recognition of the fact that any adequate illumination would always glare the opposing driver, and that the only way to meet the situation would be to provide proper illumination when there is a clear road ahead, with provision to depress the beam when another car was being met and passed. Regulating authorities in the Middle West permitted these systems to be used substantially as intended, which resulted in considerable improvement in the situation for drivers equipped with such lamps, and for about ten years all cars produced have been so equipped. Unfortunately the regulating authorities in all states have not seen fit to permit the lamps to be used as intended and have insisted on aiming the upper beam so that this would not cause glare. The double beam system has never had a chance to show what can be done. More recently, systems having passing beams have been developed in which provision has been made to depress or divert only the light which will glare the oncoming driver, light being left in the right of the beam to illuminate the edge of the road along which the car is proceeding. Traffic conditions near our larger cities are such that a driver is frequently compelled to go for miles without an opportunity to use his driving beam and without improved lighting along the edge of the road the driver is unable to take proper account of pedestrians and other objects close to the roadway.

There are many more pedestrians along the roadway than in the past. This is partly the result of the building along improved roads, particularly near cities. This seems to be the place to urge that steps be taken to educate pedestrians regarding the limitations on the driver's vision during bad weather, road, and traffic conditions. It does not seem possible that as many pedestrians would walk as they do if they understood the whole situation. We see some pedestrians carry flash lights or lanterns. Possibly the rest of them could be educated to the advantage of calling attention to their presence. Reflex buttons on belts or canes would be of assistance.

Regulations imposed by the officials, and legal restrictions resulting from the activity of these officials have hampered lighting development in other ways, even when there are logical reasons for the regulations. The laws of a very large number of states put a definite limit on the light flux to be used by the lighting engineer. There was a great deal of justified complaint because of the complication of the devices provided for adjusting the bulb in the reflector. The public did not use these very successfully. The lighting engineers developed fixed focus systems which made focusing mechanisms unnecessary. The first fixed focus systems required longer focal length reflectors, which reduced the percentage of the light flux available for road illumination. The engineer was not permitted to increase the bulb candle power to compensate. A regulation adopted with the very good intention of preventing placing excessive candle power bulbs in units designed for smaller candle power had the quite unintended result of placing a very undesirable limitation upon good design, and design intended to remove one of the most annoying faults of the then existing equipment.

Should not the specifications be directed at the result, not the means for reaching the result – at the light distribution in the beam, and not the bulb candle power? Fortunately a new form of fixed focus design called the prefocus was later developed which permitted the focal length to be made as short as with the use of focusing mechanisms.

There has been also some reduction in useful light flux available for the road, as a result of modern styling of automobiles which has made it necessary to use smaller headlamps in order to conform to other dimensions on the car. The designers should have an opportunity to use a smaller headlamp if such a headlamp will reduce wind resistance and will give an improved result from the esthetic standpoint. No one would advocate mounting on one of our larger cars today, the headlamp used in 1920 on the same type of car. If the purchaser of the car wishes to have a lamp which conforms to the rest of the car, even if this does waste some of the light flux, he should have the privilege of paying for it by the use of a little more current.

I see no hope of making any real improvement in the lighting situation as it affects safety, except by recognizing that the conditions have greatly changed since specifications were developed which came from the effort to secure illumination and prevent glare by the aiming of a single pair of fixed beam headlamps. Night traffic has tremendously increased. Speeds are very much higher than they were in 1920. The driver's eye is from 3 to 5 in. lower than it was, even ten years ago, making it much more vulnerable to glare. We need more light for seeing than the old specifications permitted, and there is more opportunity for glare because of the lower position of the driver's eye and the greater traffic density. Our only chance outside of the introduction of radically new inventions, is to recognize that the opposing views as to what constitutes good headlighting, simply expressed as demand for illumination vs. demand for glare elimination comes from the fact that we have two fundamentally different conditions to meet, and that we must have really two lighting systems to meet these conditions. A very large percentage of the drivers in the Middle West use these systems as intended without any compulsion from the police. It seems reasonable that if the manufacturer and the authorities both do their part that the drivers in all sections of the country can be educated to the necessity and advantage of using such systems.

Proposals which are very interesting from the technical standpoint have been made to use polarized light as a means of providing illumination without glare. A demonstration of

this was made at the 1936 S.A.E. Semi-Annual Meeting. The writer has not yet seen any combination of materials which has successfully met the tests which a commercial system must meet. Furthermore, if perfect materials should be available, it would be necessary to make such extensive alterations in the electrical system that great difficulty would be met in getting such equipment installed upon the older cars in service, since the cost of the equipment for replacement would be too high relative to the sales value of the old cars. This is because many times as much light flux will be required to provide the same seeing as at present, requiring new generators, regulators, and connectors, and possibly new batteries. The advantages of the system could not be obtained, unless substantially all cars are so equipped, or cars not so equipped are compelled to have their lamps so aimed as to avoid glare under any conditions. There are too many old cars in service to make it easy to apply such regulations, particularly since the cars with polarized light would be excessively glaring to those without filters.

It has also been proposed to use color filters, so arranged that a car using red lights with the driver looking through a red transparent filter would meet cars with blue equipment. With perfect filters, this system could be as efficient as the polarized light scheme. The abnormal colors would not be too desirable, but would be quite endurable as compared to the glare. The two-color scheme would also require two complete systems of filters for projecting light and for vision on each car, and an exercise of the driver's will to make the change over.

There is likely to be considerable delay before any system requiring such radical changes will be commercially available. Meanwhile, we should do everything possible to improve results obtainable with ordinary light.

The addition of reflex markers or buttons at the rear of cars is a step in the right direction. The use of these in highway markers is to be commended. Apparently the opportunity is open for more extended use of the same principle. These reflectors could be so mounted as to be visible from the side of motor cars and other vehicles likely to be used on the highway or crossing the vehicle highway. While it may prove very difficult to persuade the railways to mount these on the sides of rolling stock to give warning against dark obstructions which might move across or be left across the highway, still it might be possible to mount these at railway crossings in such a way that the running gear of a moving car would cause them to give the appearance of winkers, and thereby give a useful warning.

Brakes

Important questions relative to brakes are –

1. Distribution of braking – front and rear
2. Accuracy and uniformity of control
3. Durability

So much discussion has dealt with bringing the car to a complete stop that there appears to be a tendency to forget that brakes are used to make minor adjustments in car speed many times as often as they are used to stop. The tendency is to over-emphasize maximum possible deceleration. This is probably better, however, than a tendency which existed when front brakes were first applied, to insist that the front wheels must never slide before the rears, even under extremely slippery road conditions. This insistence resulted in some cars having only 35 per cent of the braking in front. The rear wheels would naturally slide in any attempt to make a quick stop. The skilled driver could keep such a car aimed somewhere near straight ahead during the rear wheel skid. With more front braking the unskilled driver would also go straight ahead while sliding the front wheels and sometimes with slightly more deceleration. It was undoubtedly very good judgment, however, to avoid too much front braking while the public was becoming familiar with the characteristics of cars with brakes on all wheels.

Decision as to the proper proportions involve so many considerations that it has been decided to state the writer's personal opinion as such, with what seem to be the reasons justifying this, with the full understanding that others will have different opinions and good reasons therefor. This personal opinion is that the brakes should be proportioned so that all four wheels begin to slide simultaneously on a pavement having a fairly low coefficient of friction. This means approximately fifty-fifty distribution when the rear end is slightly heavier than the front, or slightly more braking in front than behind when the weight distribution is even.

This proportion will give the maximum possible deceleration when the pavement is somewhat slippery and the stopping distance is greater than normal. It will avoid making a bad situation worse.

Brakes must be used from time to time on curves. When this situation arises, it is desirable to have the maximum possible available deceleration that can be used in a curve. If either end of the car is braked more than the other, cornering force will be lost first by that end of the vehicle. Loss of cornering force at either end will prevent continuing on the road in the curve. The cornering force is the side thrust developed by the tires.

Equal distribution of braking will tend to make the wear approximately equal front and rear. Overall service conditions are likely to be better.

The maximum dry pavement deceleration obtainable with heavy front-wheel braking cannot be used with the passenger's consent, except as an alternative to a crash. Unnecessary panic braking will cause more havoc if excessive front-wheel braking is available.

A certain type of driver will frequently overdrive his brakes just as he overdrives his headlamps. The more braking he has available under the most favorable conditions, the more likely he is to get into trouble under bad conditions.

Any specifications covering brake performance should include an allowance for a reasonable amount of deterioration. It is recommended that no specifications be adopted which require too high a percentage of that available from a 50-50 brake distribution for a 50-50 weight distribution. If the requirements are too close to the above, there will be undue pressure on the designer to put too much braking in front, so as to have some cushion between the maximum available and the specification.

Accurate control of the brakes requires that the deceleration be very closely proportional to the pedal pressure over the whole range, and that this proportionality be maintained under all conditions of weather and history of brake usage. We do not always accomplish this today. Many brakes tend to become sensitive as they wear, that is the ratio between decleration and pedal pressure gets so high that it is difficult to avoid excessive brake application, particularly when the need for braking appears suddenly. Usually when brakes have thus become sensitive, the sensitivity varies greatly with conditions, and after a car wash, or after standing over night it is practically impossible to obtain the type of braking needed to handle a car safely on a slippery pavement. The usual cause for this condition is the use of too much self-servo action in the shoes themselves, as has been explained in earlier papers before this Society. Present day materials seem to make it possible to avoid excessive self-servo action without unreasonable pedal pressure for our lighter cars. Where very low pedal pressures seem commercially necessary in heavier cars it would seem advisable to use external servos

or boosters rather than use designs that become unduly sensitive with wear. Improvements in materials may assist in the future.

Present day brakes give trouble from fading or decrease in the ratio between deceleration and pedal pressure during the course of a single brake application. Since heating seems to reduce the coefficient of friction for the combinations of materials which are available for brakes, a certain amount of fading is certain to be present. The engineer now knows how to prevent that dangerous degree of fading which resulted in brakes which could not be made to give reasonable deceleration with any pedal pressure which the driver could apply. This acute form was usually due to excessive drum expansion, and shoe curling resulting from heating. Too much self-servo action also frequently contributed, since the loss in coefficient of friction of the lining reduced the braking much faster than the coefficient decreased. All of these effects could be reduced by greater attention to cooling.

Long life and infrequent need of adjustment are highly essential qualities, but it is not possible to draw any simple specifications for these at the present time. The brakes must meet emergency conditions without permanent damage, but any attempt to specify unreasonable tests is likely to have an unfortunate effect upon production design. Thus it is perfectly reasonable to specify that the brakes must be able to stop the car at the maximum rate possible from the maximum speed without doing any permanent harm, but it is not reasonable to expect the brakes to withstand a series of such tests without any normal running intervening. This maximum deceleration test from maximum speed will certainly do some damage to lining and drum surfaces with present-day materials. If the test is repeated without the benefit of the polishing action of normal moderate brake application, destruction is certain, and the number of runs to complete failure has no definite relation to the merit of the design. The brakes should be capable of holding the speed of the car to a reasonable value down the longest mountain grade likely to be met, but it should be remembered that in service the brakes will not be needed again until after another climb, and abnormal brake dragging tests, like abnormal stopping tests, do not necessarily give any real gage of the merit of a given design.

A large possible amount of service between adjustments will contribute to safety by permitting extended use of a satisfactory condition once obtained. This means avoiding too much servo action in the shoes, and too much mechanical advantage between pedal and shoe. Attempts to obtain too soft a brake pedal without the aid of an external servo are in the wrong direction, particularly when this effort is made on a heavy car.

Comparisons of the merits of different designs by comparing the total area of the lining or the lining area per unit weight of vehicle are valid only for the same type of brakes using the same design constants, and for vehicles capable of the same maximum speed. Specifications cannot reasonably be based on these factors. Any designs which show abnormally low amounts of lining, however, must be considered as demanding tests to prove that the reasonable requirements are met, either by skillful use of materials, or by better materials than the average.

The engineer can look upon the brake situation as being under reasonably good control. Brake design has managed to keep ahead of the difficulties produced by increased car speed and reduced space available. The conservative driver is very much better off than he was a few years ago. Also there is reason to hope that we have about reached the limit of increase in car speeds and in shrinkage of wheel diameter, so that the effort which formerly went to meet radical increases in requirements, can go into refinement of detail.

Driver Vision

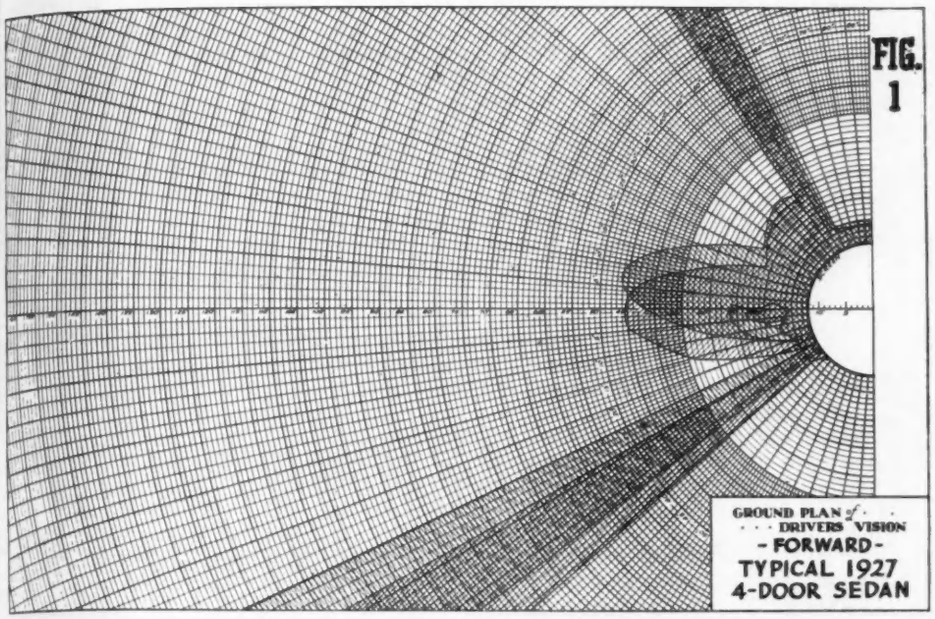
There is considerable complaint against present models because of the driver's inability to see enough of the landscape. This is another case where there is a chance to say the automobile designer has not done a perfect job. He is under pressure to change, but at the same time no steps will be taken toward important sources of the same trouble. Thus the point is stressed that the low roof combined with the rather large fillets on the front corner post, cut off vision to the right when a car is leaving a crowned road and about to enter a stream of traffic. The driver can see if he will make a small body movement so as to bring his eye into a new position. The tendency in style in ladies' hats seems at the moment to be toward wide and floppy brims. One of these on a front seat passenger will cut off vision to a much greater degree than any restriction due to body design.

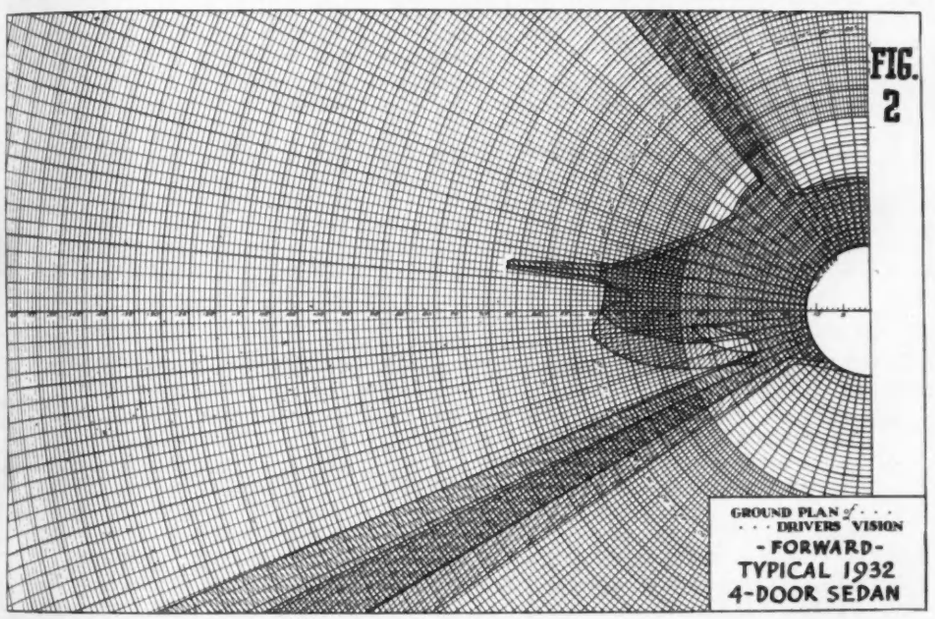
Limitations on vision, as these appear in passenger cars, are really contributions to driver fatigue. These are, of course, undesirable and sources of annoyance to the careful driver—not direct or necessary causes of danger. In so far as these restrictions to vision have come from lowering the passenger and lowering the top, the limitations have been an undesirable by-product of lowering the center of gravity which contributes to safety against overturning, and lowering the projected area of the vehicle which contributes to fuel economy. The fillet in the corner post, the undesirably large size of the corner post, and the rounded top which cuts down vision to the side, all contribute to the strength of the body structure against the forces effective if the car is rolled over.

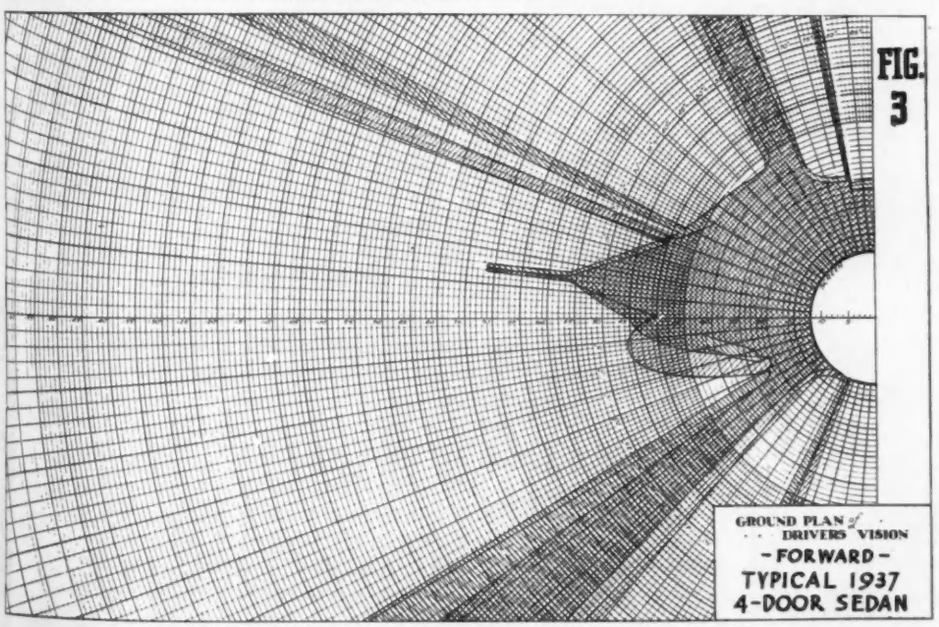
Any engineering design is well known to be a compromise. In this case the compromises which have made contributions of admitted value in certain directions, have brought factors of reduced value in another direction. We must not overstress these new disadvantages too much unless we are willing to give up the advantages. The point must also be made that whatever the faults with respect to driver vision may be, the road vision is adequate when the car is moving along the highway at any usual driving speed. The driver is not compelled to make any special motions to secure adequate vision, except when about to enter a traffic lane from a standstill or very low-speed condition, where the driver does not have the aid of very recent vision of the conditions to be met.

A complete survey of the situation for all cars would result in a volume of diagrams. Charts are here presented showing the change in vision on the assumption that the point between the driver's eyes remains fixed for a 1927, a 1932, and for 1937 passenger car of the same make, which may be said to represent the general trend. Figs. 1, 2, and 3, show the driver's vision of the road. Lowering his eyes has increased the area directly in front of the vehicle which is masked by the hood. Figs. 4, 5, and 6, show the internal area of a cylinder of 12 ft. radius centered between the driver's eyes which may be seen as the driver's head is rotated. Doubly shaded areas on both sets of charts are invisible to either eye. While these charts show reductions, they also show that small movements of the driver's head will render visible any masked areas.

Rear vision has also been reduced as shown in the charts showing the cylindrical area visible, but is still much better than in the delivery wagon which uses the passenger-car chassis, and which holds its own with passenger-car traffic. External rear vision mirrors, a suggested mounting being on the front-door hinge, have been proposed to give vision of the blind spot at the left rear quarter, which may be occupied by a vehicle whose driver intends to pass. Such mirrors would







be of real assistance to the careful driver, but since trouble during passing is likely to come from the more careless driver, their use might not affect the accident statistics to any appreciable extent. I should personally like to see them tried on the convertible, which is a type having rear vision nearly as bad as the old style open car with the top and side curtains in use, and which represented a considerable percentage of the vehicles in use not so long ago.

The wings in the front windows to promote ventilation have been criticized as reducing vision, particularly on rainy nights. If these wings are swung outwards for ventilation when driving in the rain, conditions are worse than when the old-style windows are in use as long as the rain falls straight down. Whenever there is any wind, all windows are soon smeared unless the vehicle maintains the same angle to the wind, and a left turn against oncoming traffic at night requires that the driver use extreme care, and move his head so he can see to the left through the cleared portion of the windshield in any car. The ventilating wing may be turned at any angle such that comparatively clear binocular vision is obtained without lowering the whole window as is necessary in the old-style window control. The condition is undesirable on any car, but the new design is not necessarily worse than the old for the driver willing to take precautions.

Windshield wipers do not clear as large a part of the windshield area as is desirable. Furthermore, they are nearly useless just after a rain has ceased and the car is moving over a dirty road and subject to the splash from other vehicles, but without the benefit of fresh rain to help remove the mud.

The old nearly vertical windshield protected by a visor, usually had some clear areas when starting up unless the storm was driving straight on. We have slanted the windshield and removed the visor to reduce wind resistance and must clean the windshield before starting out after any snow or sleet.

We have provided double windshield cleaners as standard equipment on the medium-priced cars, and made them available as special equipment on the low-priced ones, which has assisted driving in bad weather conditions. We still need, in my opinion, to make available boosters or windshield cleaners operated by other sources of power than engine vacuum, so that the windshield wipers, at least at the owner's option, may be made to work with the throttle nearly open. The drivers of these cars insist on keeping up with, not to say passing, the more expensive cars with more adequate equipment, regardless of weather.

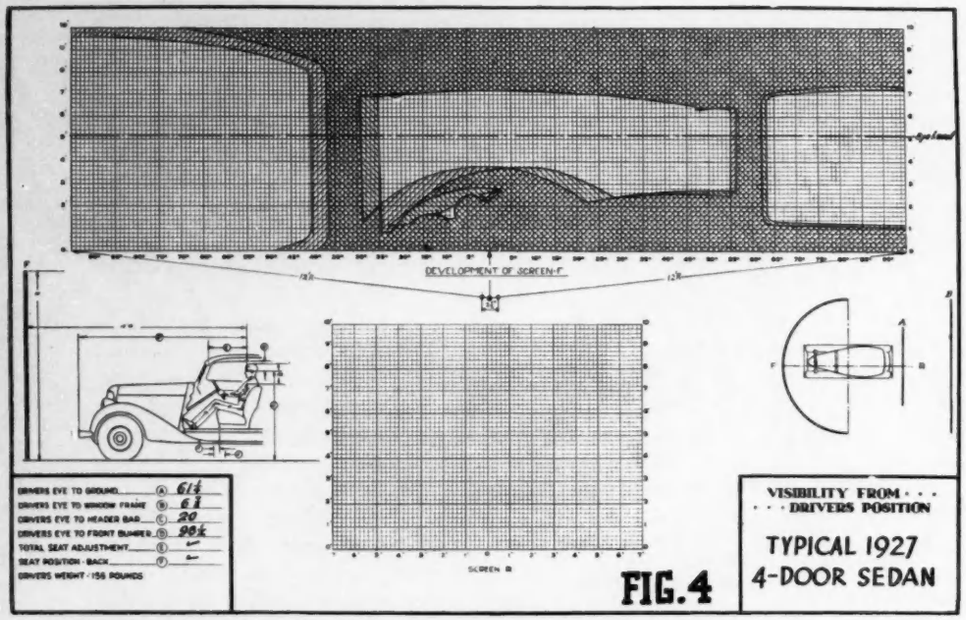


FIG. 4

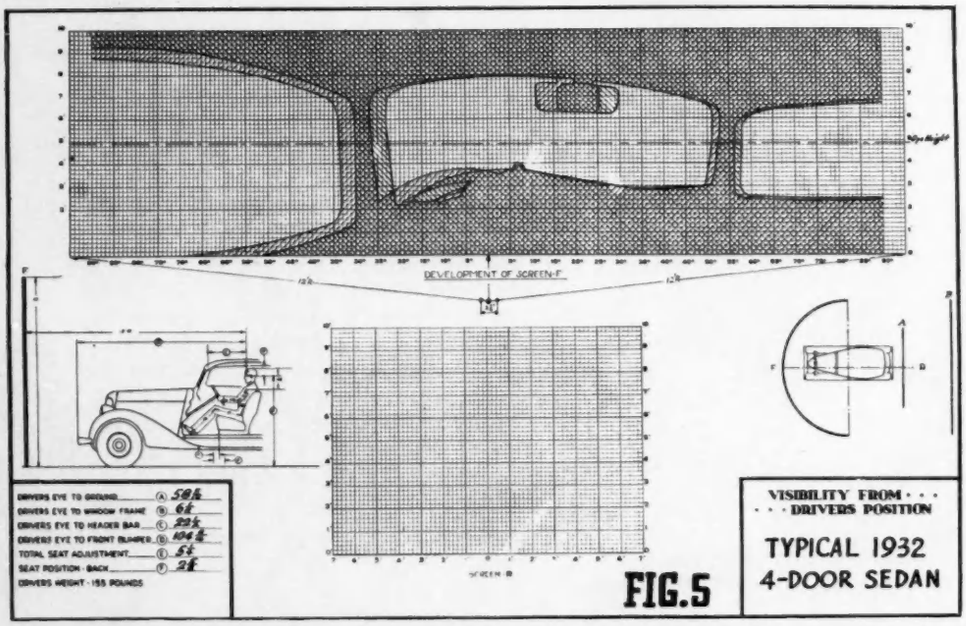


FIG. 5

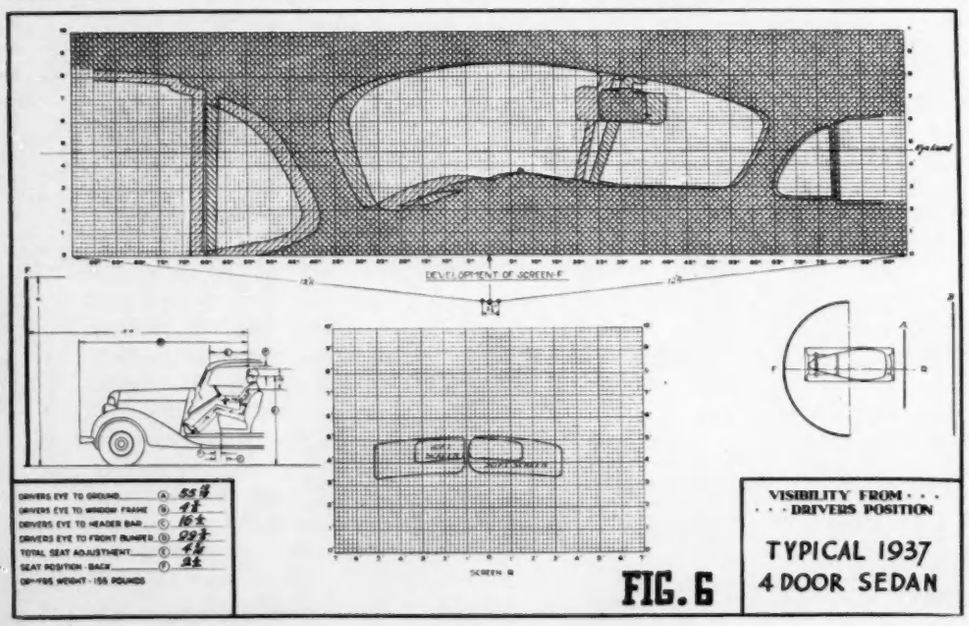


FIG. 6

We have finally provided equipment which may be installed on any car in the shape of car heaters and a fan which will handle sleet and fog on windshields, and are beginning to make at least partial provision for defrosting as standard equipment. Under bad weather conditions the vision available to the driver is much greater than in the past.

It does not seem likely that the passenger's eyes or the roof above his head will be lowered much farther. Attempts to reduce wind resistance and lower center of gravity any farther are probably stopped by the impossibility of compressing the passenger's body. We may expect conditions to be better rather than worse in the future.

In closing this part of the discussion, the observation seems pertinent that in so far as reductions in vision have reduced safety, this reduction has come principally because in making certain improvements we have placed the driver in a car providing slightly reduced vision without calling his attention to it. Psychologists have pointed out that man will adjust himself easier to a conspicuous and radical change in environment than he will to a minor and scarcely noticed one. It is not too late to repair this fault. The average buyer of the 1938 car will be changing not from a 1937 car, which may provide less vision than the new car, but from an older model which might have better vision. If we show him exactly where his vision may have been reduced, he is less likely to assume that he is seeing better than he really is, and no danger factor will be introduced.

It may be true that no driver is really prevented from seeing whatever part of his surroundings that must be seen, and that such restrictions as exist have come as the result of improvements which in themselves contribute to safety, but in my own opinion, it is necessary as a matter of good public relations to make obvious improvements in the situation. We should make an immediate start along this line since so much time is required to make changes in matters involving body stampings.

Crash Dangers

Cars are built and are purchased with the primary intention of using them to carry persons along the road. Certain advertising stunts involving rolling cars over on the level or down hill do not change this fundamental fact. There is a certain contribution to safety in constructing the top so it will not be crushed down upon the passengers in case the car turns over. However, the public is just as conscious as we are that an unfortunately large percentage of our products

sooner or later becomes involved in a collision with other vehicles or with stationary objects, in which the passengers are thrown out of their seats, with more or less serious injury. The public has already been aroused once to the point of endorsing legislation in an attempt to deal with this situation, and has required the use of so-called safety glass. It is altogether likely that other legislation may ultimately be enacted unless the industry obtains the credit for doing what may reasonably be done to improve the situation. Some surgeons are already presenting the results of their experience, and these discussions are coming to the attention of safety bodies. This contribution is certainly to be welcomed, and it is to be hoped that steps will be taken to get all of the facts regarding all of the conditions involved in accidents, so that the expedients most likely to be generally effective can be determined.

Safety glass has been accused by some surgeons of causing more damage in the average collision than would have been caused if the simple plate glass had been retained. These surgeons say that the percentage of fatal skull fractures is increased, and that the damage done by broken glass where the safety glass is shattered into fine splinters as occurs in severe crash conditions, is on the average worse than the damage done by the larger pieces of plate glass. It is to be hoped that steps will be taken to secure adequate information with regard to this.

We undoubtedly have many more crashes where the change in velocity is not enough to throw the passenger into or through the windshield, but where disfiguring or crippling injuries result, than where some one is thrown against the windshield itself. In these less violent shocks the passengers frequently pivot about the hip joint and their faces are thrown against the cowl board or the back of the front seat. Proper padding of the cowl board and the removal of all projecting knobs and handles have been advocated to improve this situation, and some steps in this direction have already been made.

The driver seated behind the wheel very frequently escapes damage in minor crashes, but in more severe ones, he is also damaged by the cowl and by the steering wheel or by the mast jacket. Since the driver's feet are usually on the pedals at the time of a crash, his knees are more apt to strike the lower edge of the cowl board than is the case with the passengers. Knee injuries are particularly difficult to handle, and require a long time for complete recovery where this can be accomplished.

An example of the difficulty of arriving at any agreed facts in this situation is the fact that two different persons whose position and experience require their opinions to be carefully considered, have made absolutely opposed statements in regard to the steering wheel with wire spokes, which is finding considerable use at the present time. One prominent engineer states he wouldn't have one on his own car, while a safety investigator states it is a real improvement. Possibly these persons have seen different accidents where the conditions were sufficiently different to give radically different results. The elastically mounted rim on the wire-spoke wheel would not deliver as hard a stomach blow as a more rigid wheel, and if strong enough to keep the driver from crashing against the end of the mast jacket, might prevent a serious injury under certain conditions of impact. If the shock is great enough to force the driver's chest against the mast jacket end, serious injuries are almost certain.

When collisions occur while vehicles are skidding, violent side shocks frequently occur, and pelvic injuries frequently occur to the rear seat passengers. Folding arm rests in the center of the rear seats would help in some cases where only two passengers are present.

It seems obvious that in many cases safety belts and helmets as used in aircraft would accomplish more in protecting passengers than modifications of the car structure itself which are likely to be practical, but it is very doubtful if the public could be persuaded to use them even if provided.

The subject of modification of car design to reduce personal injury to passengers in crashes seems to be one requiring considerable additional careful investigation. This investigation should be very broadly supported, and would have to be very broad in scope. It does not seem to be one in which the greatest benefit possible could be obtained as the result of the efforts of any single manufacturing group.

Complaints are made regarding unnecessary damage to persons outside the vehicle by our present designs. The surgeons have presented far too many cases of tragic injury from door handles and sharp radiator ornaments. Current designs are very much better than those of a few years ago with respect to both of these features. We must avoid any backward step.

The bumper position is claimed to be such that injuries to pedestrians' knees and fractures just below the knees which are difficult to treat, are more numerous than would be the case if the bumper were raised. We have here a very serious practical difficulty in dealing with the millions of cars on the road carrying the old design, assuming that the proposed change is desirable. There is real reason to doubt whether the overall change in serious accidents would be in the desired direction. This proposal seems to be one to be given special study before any action is taken.

Fatigue Factors

A tired driver has slower reactions than one who feels fresh, and what is much more important is much less apt to give keen continuous attention to driving. A very tired driver will sometimes permit his attention to be diverted by new irritations which may be supplied by passengers or other drivers. Intelligent attention to driving is the very first requisite to safety. Driver fatigue is increased not only by any increase in the effort required to control the car, but also by any source of irritation, even if the irritation may seem quite endurable at the beginning of a trip. Bad ventilation may have a particularly bad effect, particularly if any noxious gases are present. A great deal of driving today is done on schedule to the extent that any driving planned is usually completed. The driver does not ordinarily stop when he feels tired. The more comfortable the driver is in the car, the safer the driving, all things being equal. A certain percentage of the night driving accidents are undoubtedly due to excessive fatigue, most of which are in no way due to the vehicle itself.

The noise and vibration in the modern car are much less than those existing in older models at the same speed, and many cars give the impression of being much quieter at 60 m.p.h. than the same make was at 35 m.p.h. a very few years ago. In fact, noise and vibration reduction has been carried so far that there is complaint from some quarters that we have taken away from the driver the influences he formerly more or less unconsciously relied upon to estimate the speed of the car. In any event, it does not seem possible to improve safety conditions to any marked degree by further reductions in noise and vibration.

One exception to the above relates to ventilation. It is not yet possible to secure adequate ventilation on a really warm day, particularly at high speeds, without excessive windage noise under at least some conditions of relative direction of car and wind. We still have something to do along this line, but again it is doubtful if the disturbance is enough to affect safety to any important extent.

Eye strain is particularly trying to many people, particularly that caused by undue depression of a headlight beam with a sharp cut-off. This strain exists at speeds low enough so that clear vision exists to a distance beyond that required to stop.

The adjustable front seat has assisted greatly in reducing driver fatigue. This not only permits meeting the driver's desires when he starts a trip, but provides opportunity for a change in position during a trip which in itself is frequently a source of considerable relief. Vertical as well as horizontal adjustment tends greatly to reduce fatigue of short-waisted people in our present cars with high cowls. The adjustment of the front seat is sometimes a compromise between the desires of the driver and of the front-seat passenger. Separate adjustment of the front seats would impove this situation. It is believed that most persons would be more comfortable in seats that permitted more nearly the position assumed in an office chair. Unfortunately, the presence of effective four-wheel brakes makes such a position precarious to anyone not provided with a steering wheel or the equivalent on which to brace himself in time of need.

The effort required to control a modern car in good condition is now so small as not to be a real source of fatigue. In fact we have reduced this effort so much that there is sometimes a tendency to over-control on the part of drivers of our medium-weight, medium-priced cars, when first taking these over after driving older models. In families having two cars, too soft controls on the brakes are likely to result in over-controlling, especially when one of the family cars does not have nearly so soft a brake pedal. Possibly the one exception to the general tendency to reduce control effort is the too frequent existence of too high pedal pressure on the throttle control. This is particularly trying to the ladies, whose shoes have not been too much favored by our usual designs.

Bad ventilation contributes to fatigue, especially if conditions are such that the driver must choose between bad air or drafts against the side of his head and neck. We are much better than we were, but improvement is hoped for.

One accusation brought against present designs of cars is that our ventilating systems have so increased the suction on the leakages into the body that we are poisoning all of our customers with carbon monoxide from the exhaust. Mention of carbon monoxide poisoning is included here since the initial effects of mild poisoning are similar to those of fatigue. I am at liberty to state on the basis of tests made by the staffs of General Motors Research and Proving Ground organizations, that these accusations have been proved to be unfounded. The tests covered the full range of 1936 model cars in important production with added tests on certain older models. Opening side windows on any car produces a small depression within the body, but the cars with ventilating wings in the window do not have any appreciably greater depression than with the ordinary sliding window, even when the wings are placed at the most effective angle. Exhaust gas from the car's engine does not enter the body unless there exists the combination of leaky exhaust line or muffler and openings in the floors.

Present-day cars pick up practically all the carbon monoxide which may be present from the cars ahead and to windward, and not from their own engines. The carbon monoxide entering the car from its own engine under the most unfavorable conditions is not as great as that produced by one cigarette smoker.

We should maintain floor board and dash designs that are easy to keep in a gas tight condition. Second and third owners of our cars would be assisted if we should use improved materials in our exhaust connections and mufflers. We must also recognize that in lowering the floor as far as we have, we have made the exhaust system more liable to damage on vacation and hunting trips into the woods. This is another reason for continuing with good design in the floors.

Obstructions to vision are, in my opinion, a source of driver fatigue, by imposing a mental burden upon the driver to remember to move so he can be sure of seeing. The motion itself is likely to cause no real fatigue or discomfort, since the average man is sufficiently restless to require opportunity to move about from time to time.

Boredom frequently produces the same dangerous effect as fatigue, a tendency toward permitting attention to wander. Many drivers feel bored when driving over a monotonous road, particularly when the road is in excellent condition, and presents no driving problems. Driving at night on a straight smooth road through the illuminated tunnel effect is particularly likely to develop boredom, which may easily be complicated by real fatigue. It seems reasonable that under such circumstance a car radio would reduce boredom and really add to safety.

The above is offered with due humility. The problem is primarily one for the psychologist and the physiologist. The engineer can only remove or reduce sources of fatigue after information about fatigue sources has been supplied. We do not seem to have enough information about the effect of various combinations of annoyances upon people, even if some data are available as to the effect of simple disturbances, such as certain sounds and vibrations. Furthermore, the sensitivity of different individuals varies tremendously. In the absence of scientifically proved data as to sources of trouble, we can only use common sense in attacking this problem, and are not justified in claiming too much credit for the result. We are justified in believing that the overall effects of the engineers' efforts in the last few years have been toward improvement.

Roadability

The ideal car should not only require no fatiguing physical effort in operating controls, but should also avoid any fatiguing mental effort. The response to the controls should be substantially the same under all conditions of driving. We cannot secure this. Opening the throttle at 60 m.p.h. does not have the same effect as opening the throttle at 20 m.p.h. Applying the brakes in a curve causes results which are not apparent in the straightaway. A given turning of the steering wheel may have one effect at one speed and on one type of road, and a quite different effect at a different speed on a different type of road. The best that we can do is make sure that as far as conditions due to vehicle design are concerned the transition from the effect of the controls under one condition to that under another is smooth and continuously in the same direction.

The response of the engine is so certain and smooth in the modern car that we hardly give it a thought. Every driver passes other vehicles under traffic conditions where a momentary hesitation in the power supply is potentially very dangerous. Therefore, the installation of a speed-limit governor which would abruptly change the available acceleration over a narrow range of speed would introduce a new traffic hazard. It would not be safe to attempt to pass another car unless the driver first carefully checked his own speed, the space available, and the speed of approaching traffic. The sudden change brought about by the governor in the response to the pedal would be highly undesirable.

Brakes have previously been discussed, and we will merely mention in passing that great variations in response to the

application of pedal pressure, whether due to fading or due to undue sensitivity, obviously introduce control hazards.

The response to turning the steering wheel cannot be made the same for all speeds because of tire characteristics. If the wheel is turned to the locked position and the car driven very slowly, we obtain the minimum turning circle available for parking. Disaster would be certain if we would rely on obtaining this same turning circle to avoid an obstruction when going at quite moderate speeds. We would find that the turning circle would be very much larger than at the very low speed. The relation of the rear-wheel tracks to the front-wheel tracks would not be that of the Ackerman steering diagrams. The rear-wheel tracks will, under good conditions, be only slightly inside those of the front-wheel tracks, and under bad conditions might be well outside.

Space limitations prevent more than a very superficial discussion of these factors. It is to be hoped that those active in studying car handling will present complete discussions in the future.

The data presented before this Society by Captain Evans have shown that for all practical purposes a tire does not develop any side force to compel a car to make a turn until there is an angle between the path at which the tire is aimed and the path along which the tire is actually moving. Stated otherwise, the tire must drift or creep sidewise as soon as any side force is developed. Cornering force per unit of load falls off as the load is increased upon a given tire. It is therefore necessary to load the tires at the two ends of the car to substantially the same amount, and to have the ratio of weight transferred from the inner tires to the outer tires in a turn substantially the same at each end of a car where four identical tires are used if the car is to have the maximum ability to make a turn.

Using Maurice Olley's nomenclature, we will say that a car which shows a tendency to slide outside the turn in front more than at the rear as the speed increases, and which, therefore, requires an increased turning of the steering wheel as the speed increases on a curve of constant radius, is a car which understeers. If the rear end slides out farther than the front end, we say that the car oversteers. A car which oversteers will require a certain turning of the wheel to cause the car to enter a curve of a given radius, and once in the curve, if the speed is maintained, the steering angle must be reduced or the car will tend to run off the turn on the inside. This is because the outward swing of the rear end changes the angle between the steered wheels and the road. The car is, therefore, unstable. Constant skilled attention on the part of the driver will be required to keep the car in the desired path. Camber of the front wheels reduces side thrust. Therefore, independent suspensions which permit front-wheel camber changes when the car rolls in the turn tend to understeer. The conventional front suspension is rather more apt to oversteer.

Cars which had oversteer characteristics have been accepted in the past by at least a percentage of European drivers. The low power-weight ratio of European cars makes it undesirable to slow down more than is absolutely necessary in making the turn. These drivers would dash up to a turn at a speed much higher than an average American driver would attempt, rely upon the oversteer to help line up the car with the new direction of the road, and rely upon power application to control the skidding in the turn. When two-wheel brakes were in use generally, it was a well-known trick of skilled drivers to throw a car into an oversteer condition by the momentary application of the brakes in a turn, followed by a full application of power.

The angle between the car axis and the road may be changed not only by turning the steering wheel, but also by changes in wind pressure or by effect of crown angle of road. Once the car axis angle has changed, the oversteer car tends to turn farther requiring immediate steering correction. As an example of the relative roadability of the two types of cars, an extreme case might be cited. A high-priced European sport car having oversteer characteristics was driven in a gusty quartering wind on one of the four-track roads near Detroit. A skilled driver could just manage to keep the car within the space of three of the four lanes at speed. Under exactly the same conditions a low-priced American car having understeer characteristics would go straight down one lane with practically no effort on the part of the driver.

The purpose of the above discussion is to provide a basis for pointing out not only that oversteer characteristics should be avoided, but that from a safety standpoint an abrupt transition from understeer to oversteer must especially be avoided. Changing from understeer to oversteer while going around a curve, as a result of characteristics built into the car, would have very much the same effect as the use of such faulty steering geometry that the rolling of the car in the curve would cause an additional turning of the front wheels, tending to make the car turn violently under conditions where the driver is already disturbed. Obviously, a driver must be quite experienced to deal with such a situation.

We can avoid, by careful design, transitions due to built-in car characteristics. Nevertheless, we must recognize the fact that transitions do occur because of conditions outside of the designer's control. Thus, when a car is being driven in a turn, if the car is provided with a really high percentage of front wheel braking, brake application will cause the car to develop a high degree of understeer. Panic applications of the brakes can bring the understeer to a condition of no steer at all by skidding the front wheels with resulting disaster if a turn is necessary. If the brake distribution is more nearly equal, or with a high percentage of rear wheel braking, and road traction conditions permit developing weight transfer effect, a panic brake application will cause oversteer with its own possibilities of trouble. A car which normally understeers may oversteer when the rear tires are too soft, or if excessive load is placed at the rear of the car. When a rear tire blows out, the change is exceedingly abrupt.

The above discussion has mentioned curves and conditions in the curves until the impression may be left that car handling in curves is more important than it really is. The most important thing is to get a car which responds uniformly and constantly to the driver's will on what is called straight-ahead driving, where changes in road camber, road surface, angle of side winds, bring about constantly changing conditions. The car with moderate understeer will give good handling under this condition. The oversteer car requires constant and fatiguing attention from the driver, and at speed develops conditions highly dangerous to the unskilled driver. A car which gives thoroughly good handling for all conditions on the straightaway will usually be quite satisfactory in the curves.

The discussion of roadability has assumed that the spring rates at the two ends of the car and the adjustment of shock absorbers is such that there is no violent pitching of the vehicle, and that wheel hop is controlled so that the maximum effective road contact is being maintained while driving. Improvement along these lines has been so great in the last few years that there is a tendency to forget how bad an effect upon car handling can come from poor axle control.

Steering gear ratios have come in for adverse criticism. The use of low-pressure tires, combined with the need for bringing the operation of parking within the strength of the woman driver, have caused steering gear ratios to be much higher

(*Continued on page* 369)

Modern Automotive Electrical Equipment

By R. M. Critchfield
Chief Engineer, Delco-Remy Corp.

THE electrical equipment of the modern car differs from that used on early models mainly because of the increased electrical loads involved and the increased range in speeds of operation. Increased headlamp loads, ignition load, and horn load, and the introduction of certain accessories such as heaters, radios, defrosters, and so on, account for these load additions.

These changes in electrical loads and speed ranges have affected the generator requirements most seriously, resulting in enlargement of its capacity and a change in its speed-output characteristics. For this reason we will deal first with this unit of the equipment.

The Generator and Its Control

In order to bring out the progress made in the generator equipment in recent years, it may be well first to consider the equipment of some ten years ago. The electrical load of the typical 1926 car follows:

Two 21-cp. headlamp bulbs	5.0 amp.
One 3-cp. tail-lamp bulb	0.5 amp.
One 3-cp. dash-lamp bulb	0.5 amp.
Ignition	1.5 amp.
Total	7.5 amp.

Fig. 1 shows the output curve of a typical generator used at that time and found to be satisfactory, providing sufficient energy to handle the loads mentioned and replenish the energy used in starting, horn operation, and so on. The maximum speed at that time did not greatly exceed 55 m.p.h. The generator output characteristic was such as to carry the electrical load through a speed range of from 14 to 39 m.p.h. Most of the driving was within this range.

When operated in the daytime with only the ignition load, the excess generator output which had to be absorbed by the battery was not enough to cause damage to the battery. Touring speeds were in the neighborhood of 35 m.p.h., and a pronounced drooping output characteristic with the third-brush generator used resulted in a hot output of approximately 8 amp. This output gave a battery-charge rate which, although slightly high, did not constitute a serious overcharge.

Some difficulty was experienced due to high voltages during winter driving. On the other hand, cars were not driven much in the cold weather, and the arrival of spring brought relief in a natural, if not scientific, manner. In order to prevent overcharge of the battery and high voltages, some owners drove with their lights on in the daytime, but this expedient did not prove a very satisfactory method of control.

[This paper was presented at the Baltimore Section Meeting of the Society, Baltimore, Md., Mar. 4, 1937.]

Now, let us turn to the electrical load requirements of a modern car. The loads required by a typical electrical system follow:

Two 32-cp. headlamp bulbs	8.00 amp.
Two 3-cp. tail-lamp bulbs	1.00 amp.
Three 3-cp. instrument-lamp bulbs	1.50 amp.
Ignition	1.75 amp.
Heater	4.00 amp.
Radio	8.00 amp.
Total Load	24.25 amp.

As against a total load of 7.5 amp. encountered on the car of ten years ago, we now find a total load in the neighborhood of 24 amp. In fact, a survey of the loads measured on thirteen different-make 1937 model cars, showed an average of 24.34 amp. The light and ignition load averaged about 12 amp., whereas heater, radio, and defroster requirements account for the remainder.

Sales of heaters and radios for the past several years are listed as follows:

Year	Heater Sales	Radio Sales
1930	270,000	34,000
1931	330,000	108,000
1932	375,000	143,000
1933	750,000	724,000
1934	800,000	780,000
1935	1,535,000	1,100,000
1936	2,250,000	1,625,000

It will be noted from the figures listed that this additional equipment is being placed on a sufficiently high percentage of the total car production to warrant provision for adequate generator capacity as a matter of standard equipment. In addition to the added loads encountered, the modern car is also capable of being operated over the much greater speed range of approximately 10 to 75 m.p.h. Inasmuch as the third-brush generator in the past has been used almost universally on passenger cars and its output characteristic is such as to tend towards reducing the output at higher speeds, the increased speed range directly affects the available capacity of this type and, due to the high rotational speeds encountered, also presents quite a problem with reference to commutator and mechanical design.

Generators are being provided which meet these increased requirements without increasing the size, or greatly increasing the cost. This result has been accomplished by ventilation and improved brush rigging and commutators, improved methods of winding and balancing the armature, and refinements in design which result in flattening the speed-output curve. Fig. 2 shows a cross-section of a modern generator.

An inexpensive fan has been provided which operates in conjunction with the belt pulley and draws air through the opening in the commutator end frame across the armature and field windings, and out the drive end frame. In this connection, it might be observed that practically all passenger-car generators are now belt-driven. This arrangement, in addition to providing flexibility in mounting and drive ratio, results in a generator design that is adapted easily to ventilation. Reaction brush rigging is used as it has been found to give the best high-speed performance. The armature coils are form-wound, assembled in the core by special machines, and securely wedged in place. The armature and pulley assembly is dynamically balanced.

Fig. 3 shows typical speed-performance curves for this type of generator. These curves indicate a maximum hot output of 22 amp., with 18 amp. output at top speed, compared to values of 11 and 4 amp. for the older type.

Observation of this chart indicates that the normal load would be carried very nicely, and that the generator would practically balance the total load, including accessories. Although this higher output generator may be called upon to carry loads approximating its maximum capacity, it still will be operated over considerable periods of time with only the ignition load. Reference to the early-type electrical system indicates that the difference between the maximum and minimum loads amounted to only 6 amp. Now this difference exceeds 22 amp.

The handling of this additional generator capacity has introduced problems that can be understood better by presenting the results of tests made on a modern car system under various conditions of speed, temperature, load, and state of charge of the battery.

Fig. 4 shows the wiring diagram of the test set-up which consisted of regular car equipment plus meters which were arranged to read generator voltage, battery voltage, headlamp voltage, generator output, and battery current. The equipment was operated over the speed range of the car under certain specified conditions, and various readings noted from

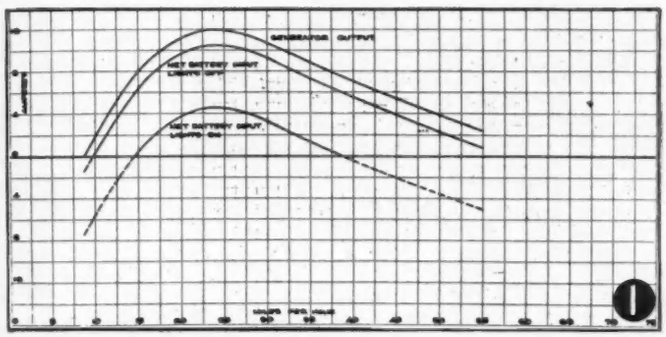


Fig. 1 – Hot Performance of 1926 Generator

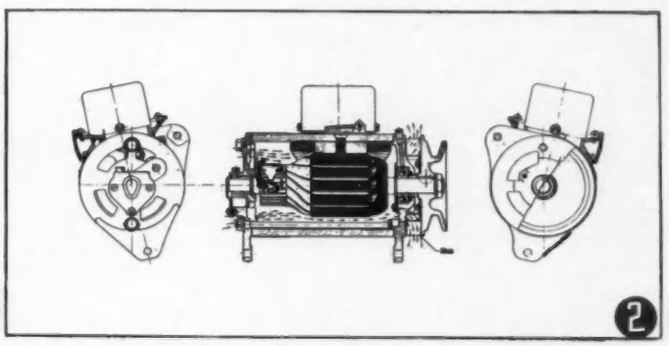


Fig. 2 – Modern Belt-Driven Generator with External Fan

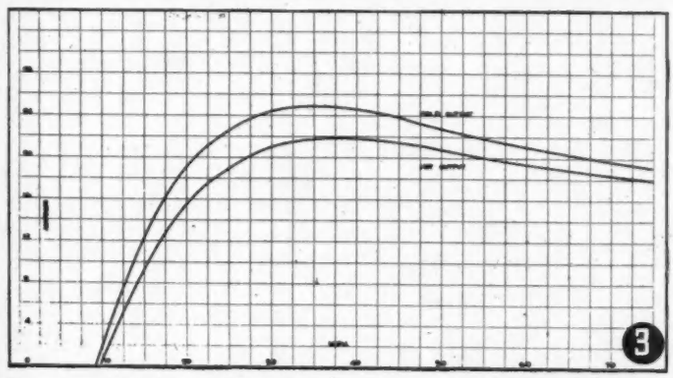


Fig. 3 – Performance Curve of Modern Third-Brush Generator

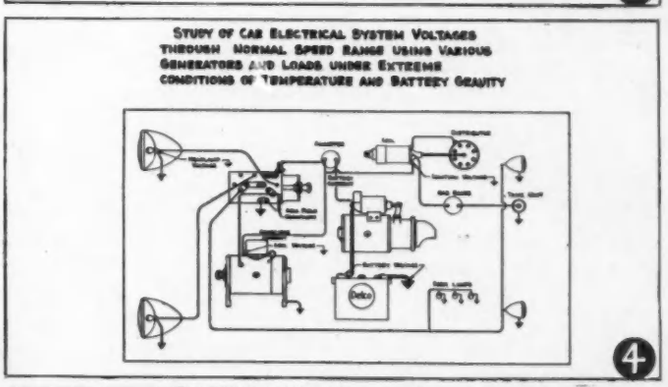

STUDY OF CAR ELECTRICAL SYSTEM VOLTAGES
THROUGH NORMAL SPEED RANGE USING VARIOUS
GENERATORS AND LOADS UNDER EXTREME
CONDITIONS OF TEMPERATURE AND BATTERY CHARGE

Fig. 4 – Schematic Diagram of Equipment and Connections Used in Determination of Car-System Voltages

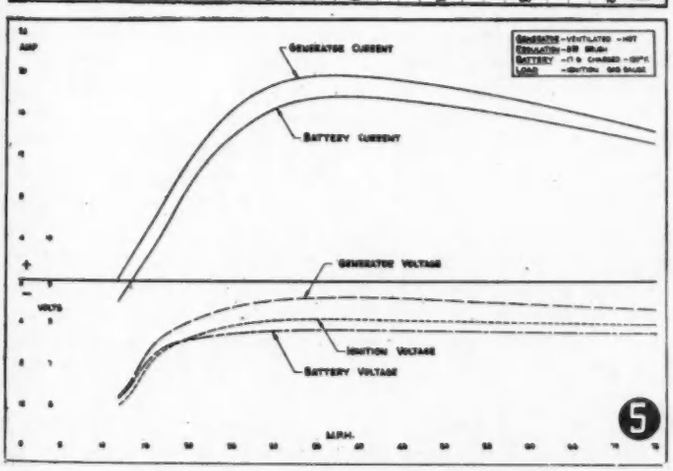


Fig 5 – Voltages and Currents for Summer Touring Conditions

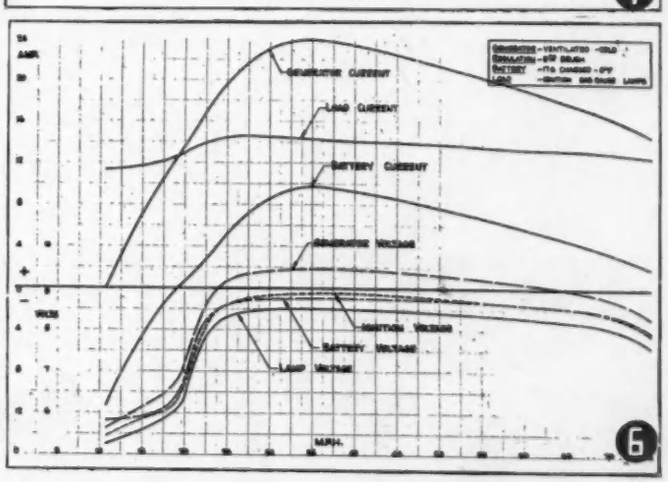


Fig. 6 – Voltages and Currents at 0 Deg. Fahr., with Battery Fully Charged and Ignition and Lighting Loads

which curves were plotted. The conditions chosen were those which would give extremes of system voltage and battery charge.

Fig. 5 gives the results with summer temperatures, battery fully charged, and ignition load only or, in other words, for summer touring conditions when the battery is most apt to be overcharged. Examination of this chart indicates that a maximum battery charge rate of approximately 18 amp. is attained. The safe continuous charge rate for the battery used in the average high-production car is probably about 10 amp. It is evident from these figures that the battery would be severely overcharged, resulting in high battery temperatures and excessive gassing with probable buckling of the plates and loss of active material.

Fig. 6 gives similar results under conditions which tend to produce high voltage in the system, namely, operation at 0 deg. fahr. temperature, battery fully charged, with ignition and lighting loads.

Voltages of the order of 8.5 to 9 are obtained and, because of the increased current consumption resulting from these higher impressed voltages, they cause severe arcing and oxidation of the ignition contacts, and considerably shorten the life of lamp bulbs and radio tubes.

The curves in Fig. 7 show the relationship between terminal voltage and life for the headlamp bulbs.

It will be noted that a bulb having a 400-hr. rated life when operated at its design voltage of 6.1, will have less than 10 hr. life when operated at 8.5 volts.

It might be argued that a larger battery would answer the problems arising from the use of higher capacity generators. A higher capacity battery would tend to absorb, without as much damage to itself, increased charge rates resulting from the use of the higher capacity generator. However, a higher capacity battery would not correct the high voltage conditions, as will be noted from examination of Fig. 8 which shows the relationship between terminal voltage and charge rate at 0 deg. fahr. for 13-low-plate and 17-high-plate batteries.

Although the 17-plate battery has a capacity approximately 50 per cent higher than that of the 13-plate unit, the curves indicate that the terminal voltage with a given charge rate is only reduced about 1 per cent. With the existing tendencies to equip present-day cars with high-capacity generators, it would appear, therefore, that some kind of voltage control is necessary.

Fig. 9 gives the schematic wiring diagram of a voltage-regulator system.

The regulator unit is essentially an electromagnet with the winding energized by the system voltage, so that the magnetic pull at the regulator armature is approximately proportional to the voltage imposed at the regulator terminals. When the voltage attains a predetermined value, the armature is attracted towards the electromagnet core, opening the contacts, thereby inserting a resistance in the generator field circuit. This resistance lowers the generator voltage and output and thus lowers the voltage on the system. The magnetic pull on the armature is thereby reduced, and a return spring causes the armature to reclose the contacts, which action restores full field strength to the generator. The generator output is thus increased, and the voltage rises until the points are opened again. Under conditions tending to produce a higher voltage than that for which the regulator is adjusted, this cycle is repeated just frequently enough to maintain the voltage at the adjusted value.

In order to determine what, if any, benefits were obtained by using voltage regulation with higher capacity generators, the tests just described were repeated with regulation added. Fig. 10 gives test results under conditions similar to those shown on Fig. 5 which were made without regulation.

The curves indicate that, under summer touring conditions with a fully charged battery, the battery input has been reduced to approximately 5 amp., or to a very safe value. This reduction in generator output when it is not needed not only protects the battery, but also greatly increases the life of the brushes and commutator.

Under conditions of operation in zero weather with a fully charged battery, when extremely high voltages would be expected, Fig. 11 shows results with voltage regulation.

Comparing these values with those shown on Fig. 6 without regulation, it will be observed that the ignition voltage has been reduced from a value of around 9 volts to a value of 7.75 volts, and the lamp-bulb voltage has been reduced from 8.5 to approximately 7.5 volts. These reductions result in considerable improvement in lamp-bulb life, and in the factor of safety in operation of the ignition system. However, in view of the fact that the ignition system is nominally rated at 6 volts and the lamp bulbs have a design voltage of 6.1 volts, the question arises as to why the regulator is not set so as to offer still greater protection to these units.

The charging characteristics of storage batteries prevent further improvement in this direction, and require that a compromise in regulated voltage be established to meet conditions encountered under various temperatures and states of charge of the battery. Fig. 12 summarizes briefly the battery characteristics which limit the benefits which can be obtained by voltage regulation.

On this chart are listed the battery terminal voltages under various rates of charge and discharge, and with the battery in both fully charged and discharged conditions at both 0 deg. fahr. and 80 deg. fahr. Reference to these curves indicates that, although a warm battery can be maintained in a charged condition with the charging voltage regulated at 7.5, a discharged battery at 0 deg. fahr. temperature would take a charge of only 3 amp. if the charge voltage were regulated to this value.

Obviously such a low charge rate would not be considered satisfactory in attempting to bring up a discharged battery. It is therefore necessary to temperature-compensate the regulator, so that a higher regulated voltage will be maintained under cold-weather conditions. Our experience indicates that a regulator set to hold 8 volts at 0 deg. fahr. provides a maximum protection for current-consuming units, and yet permits a reasonable charge rate and gravity balance for a cold battery.

In an effort to overcome the disadvantage of the drooping-speed-output characteristic of the third-brush generator and further increase capacity, a compound-field generator with improved ventilation has been developed, and is in use on some 1937 model cars. One field is connected to the third brush, the other is shunt connected. By balancing the strength of these fields to the proper degree and by improving the armature winding and magnetic circuit, a performance has been obtained as illustrated by Fig. 13.

Although using the same frame as the 1926-type generator (performance shown in Fig. 1) the hot output has been stepped up from 11 to 26 amp. and the high-speed characteristics vastly improved. This performance should be adequate to carry all normal and accessory loads under the most severe present-day conditions of driving, and experience to date has been very satisfactory.

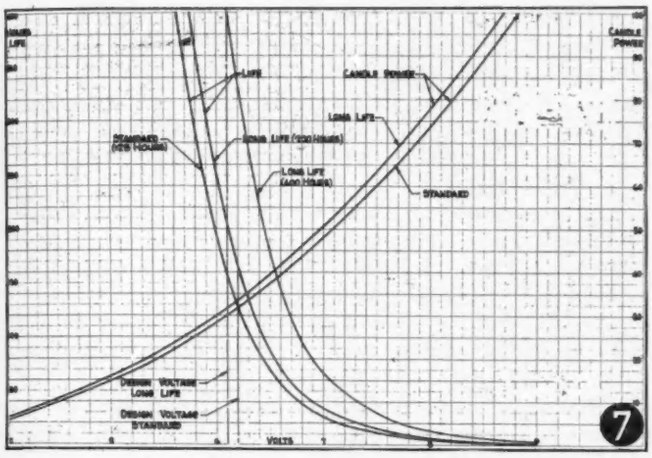
Another attempt to provide approximate voltage control at a lower cost resulted in the use of so-called light-switch control. Fig. 14 shows a diagram of this arrangement.

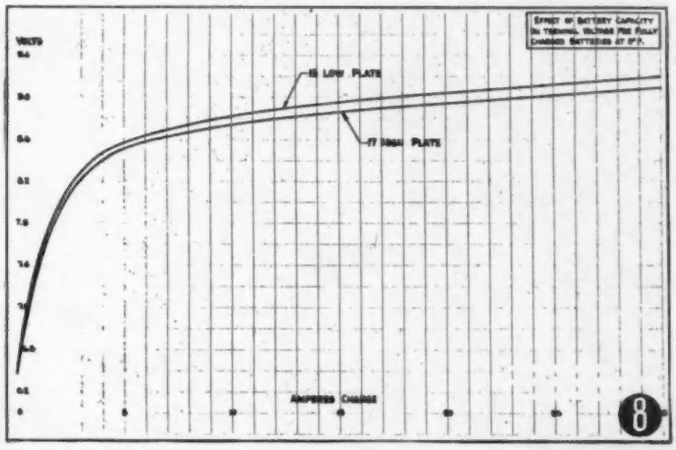
The generator field circuit is carried up to the lighting switch which is arranged so that, when the lights are turned off, a resistance is inserted in the field, thus reducing the generator output. When the lights are turned on, full elec-

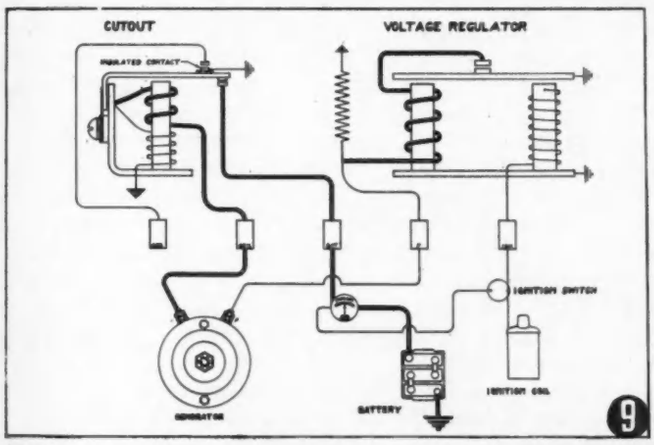
trical load is imposed on the system, the resistance is shorted out, and the generator output is stepped up to its maximum. In order to take care of the condition where it is desired to bring up a discharged battery, an extra position is provided in the switch whereby the field resistance can be shorted out without turning on the lights. This control obviously is only approximate, and is adaptable only to medium-capacity generators.

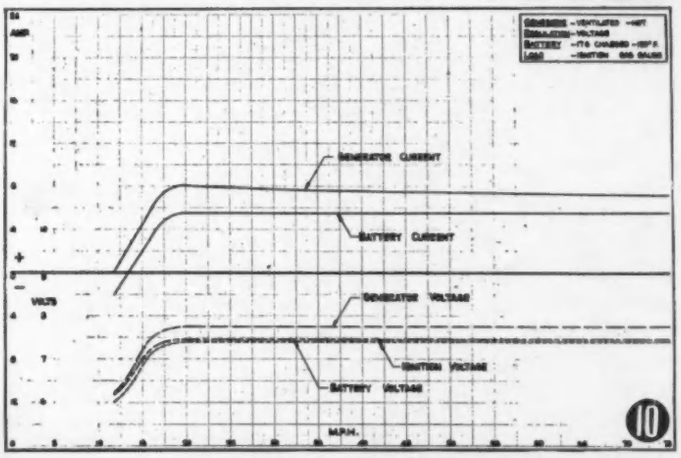
Regardless of how a third-brush type generator is modified, it still has one inherent weakness which is exemplified by the curves of Fig. 15.

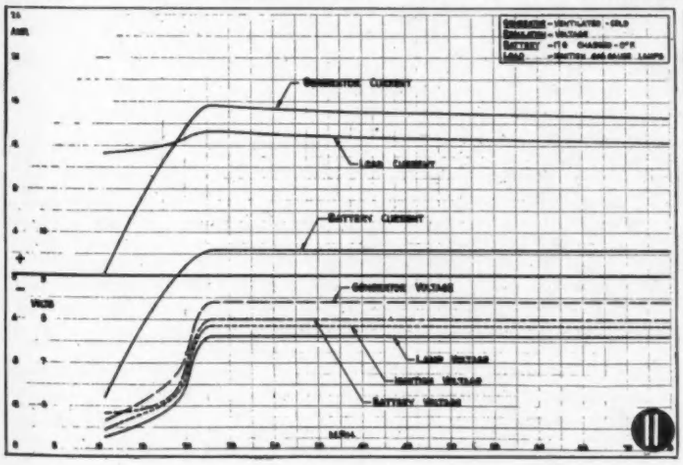
This chart shows several output curves taken on the same generator, but under conditions which gave different system voltages. It can be seen that the generator output is affected seriously by the system voltage. The generator charted has an output of approximately 26 amp. when operating in conjunction with a charged battery at 0 deg. fahr. when 10 volts are obtained at the generator terminals. The same generator, when charging a warm discharged battery, has an output of only 19 amp. Thus it can be seen that it is characteristic of this type of generator that its output is the least when it is the most needed.











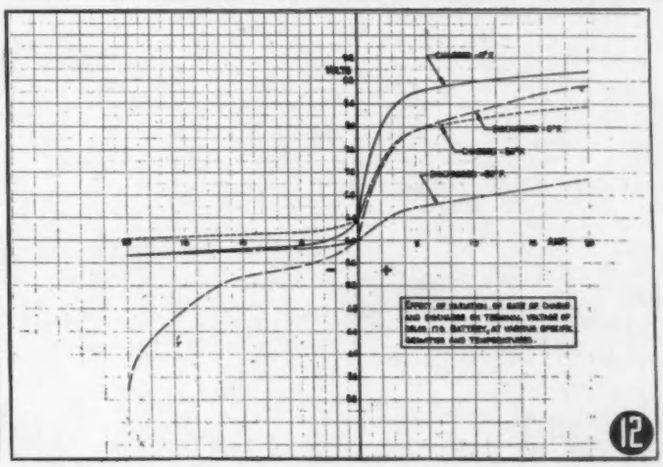


Fig. 7 – Effect of Voltage Variation on Life and Candlepower Output of Nominal 32-Cp. Lamps

Fig. 8 – Effect of Battery Capacity on Terminal Voltage for Fully Charged Batteries at 0 Deg. Fahr.

Fig. 9 – Vibrator Voltage Regulator

Fig. 10 – Voltages and Currents for Summer Touring Conditions with Voltage Regulation

Fig. 11 – Voltages and Currents at 0 Deg. Fahr. with Voltage Regulation

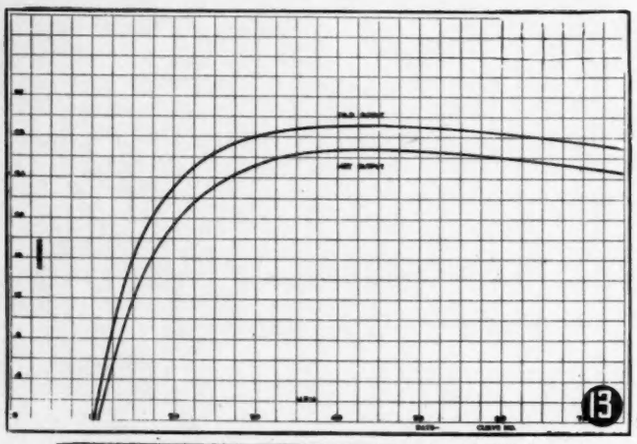
Fig. 12 – Effect of Variation of Rate of Charge and Discharge on Terminal Voltage of Delco 17G Battery at Various Specific Gravities and Temperatures

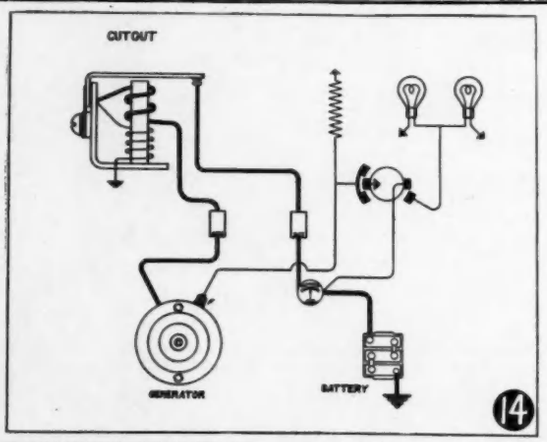
In an effort to overcome this difficulty, shunt generators provided with current and voltage regulation have been developed and are in use to some extent. Fig. 16 gives the performance of a 26-amp. passenger-car generator.

It will be noted that both the low- and high-speed performances are improved as compared with the best third-brush-type performance.

Delco-Remy production of vibrator-type regulators amounted to approximately 12,000 units in 1934, 138,000 in 1935, and 378,000 in 1936, and it is estimated that 1937 production will come well over 700,000. These figures give evidence of the rapid growth of this type of control.

In an effort to summarize the progress made during recent years on generator outputs, Fig. 17 illustrates on one sheet, plotted to the same scale, the later type generators compared to the 1926 type. The gains made in capacity are obvious.





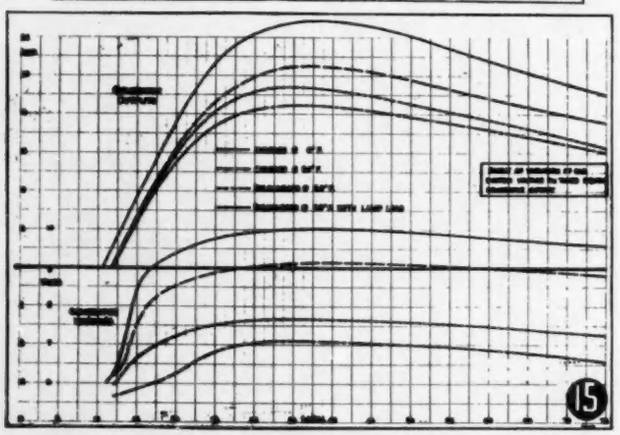


Fig. 13 – Performance Curve of Third-Brush Generator with Compound Field

Fig. 14 – Lighting-Switch Control

Fig. 15 – Effect of Variation of Car-System Voltage on Third-Brush-Generator Output

Various other generators have been developed for police car, taxicab, door-to-door delivery, tractor-trailer, city, and interurban bus service, in capacities of from 100 to 1200 watts. Fig. 18 gives the performance curve of a heavy-duty, 12-volt, city type-bus generator.

It is interesting to note the low speed at which this generator builds up to its full output. Reliability naturally is emphasized on units of this type, and many of them have established service records of well over 100,000 miles between overhauls and several hundred thousand miles of useful life.

Batteries

Although the charge characteristics of the battery were covered necessarily in connection with the generator discussion, the discharge characteristics, especially those bearing on its functions as a source of energy for the starting motor, were not discussed. These functions probably constitute its most important duties since the generating equipment cannot operate, nor can the car be used, unless the engine is started. This duty becomes more difficult as engine and battery temperatures drop, and as the battery efficiency goes down in cold weather while the engine cranking requirement is increased. The reliability of the starting system is evidenced by the fact that most present-day cars are not arranged for hand-cranking.

The battery improvement in the past several years has centered principally on meeting the increased starting-load requirements resulting from more severe cold-engine cranking loads, and an increase in the tendency to use cars in extremely cold weather. Improvements in the required battery characteristics have contributed to the solution of this problem without increase in the size of the battery container. One improvement has been the use of a greater number of thinner plates per cell. The other improvement has been in the performance of the negative plate. For example, in 1926 the 13-plate battery using 0.090 in. thick positive and negative plates was used almost universally in cars of the low-price group. A typical battery of this size was rated at about 85 amp-hr. when discharged at the 5-amp. rate. Its 20-min. rate was about 99 amp., and it would have lasted about 2 min. when discharged at a 300-amp. rate while at 0 deg. fahr. temperature.

Improved processes have raised the present-day rating of this battery to about 88 amp-hr. at the 5-amp. rate, 103 amp. for the 20-min. rate, and 2.9 min. at the 0 deg. fahr. high-discharge rating. The percentage of improvement shown is significant, especially in the case of the last rating, which is a measure of the cold-weather starting ability of the battery. This improvement has amounted to approximately 25 per cent. This improvement has been due mostly to negative-plate processing, and the comparisons are on the basis of the 13-plate cell.

However, at the present time the batteries generally used on low-priced cars are made with 15 plates per cell. The positive plates used are 0.090 in. thick and the negative, 0.070 in. thick. A variation of this arrangement has been introduced in the past few years as a low-type 17-plate battery, in an effort to maintain road clearance with lower body designs. The total plate areas, however, are about the same as the 15-plate design under consideration. The 15-plate batteries with the same case size as the 13-plate type used for comparison gains approximately 7 per cent on the amp-hr. rating, 12 per cent on the 20-min. rate, and shows approximately 15 per cent improvement on the 300-amp. discharge rate at zero.

Summarizing the results obtained with various improvements, it is believed that the present-day battery is approximately 15 per cent better in hot-weather performance and 45 per cent better in cold-weather performance than the battery

of ten years ago. The minutes rating with 300-amp. discharge at 0 deg. fahr. gives a fairly good measure of the cold-cranking ability of a battery. Fig. 19 plots this rating versus engine displacement from data taken in a recent survey of truck electrical equipment. A trend toward the use of batteries with greater cranking ability with the larger engines is noted.

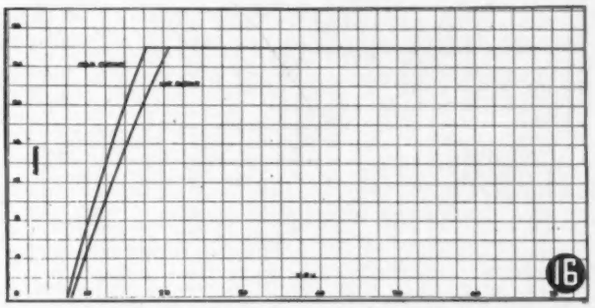
While on the subject of batteries, one item which invariably brings about low-battery complaints in mid-winter is the condition of the battery in a new car ready for delivery to the owner. Of necessity, production on batteries must start some time before they actually are needed in the car manufacturing plant. Another interval of time exists before the car is shipped since it is necessary to have enough stock on hand to insure availability of material to take care of fluctuation in the schedules. Still another period of time is encountered in shipping the car and making it ready for delivery to the owner. Any battery in standing tends to self-discharge, and the plates tend to become sulphated. Although in most cases this is not a serious matter and is not detrimental to the quality or life of the battery, it does affect what might be termed its capacity efficiency for the first several cycles of its operation. Due to the present practice of introducing new-car models in the early part of the winter, a large quantity of the cars are delivered with the batteries in a none-too-efficient initial state, at a time of the year when, unfortunately, it is necessary to use lights and heaters a greater proportion of the time, and there is a more severe demand on the battery in starting. If the owner has purchased a radio, it no doubt is used excessively, at least until the newness wears off. The combination of the heavy current requirements with the new battery, which is probably not at its best as regards efficiency, results in complaints of inability to maintain the battery in a charged condition. Experience teaches that, in general, if the battery in such instances is removed and given a thorough charge, the situation is relieved.

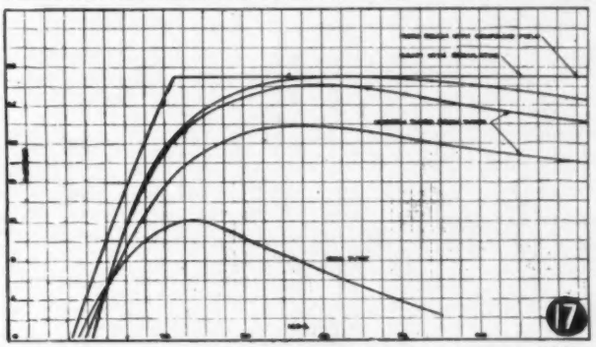
Starting Motors

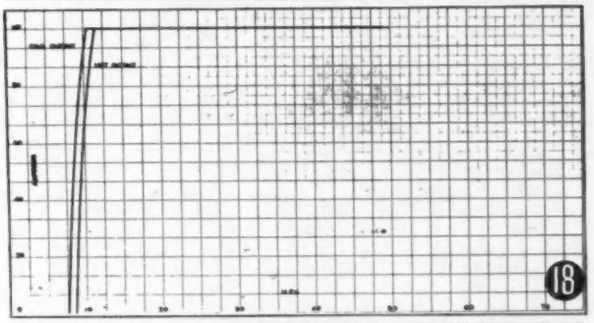
There has not been much improvement in the starting motor design itself during the past several years. Some modifications have been made in design which have resulted in minor increases in efficiency. The aggregate of these increases probably would not represent a change of over 10 per cent. As stated under the subject of batteries, the most severe conditions that a starting motor has to cope with are encountered in cranking the engine at zero temperatures. As stated also at that time, marked improvement has been made in the batteries which has contributed materially to the present improved condition.

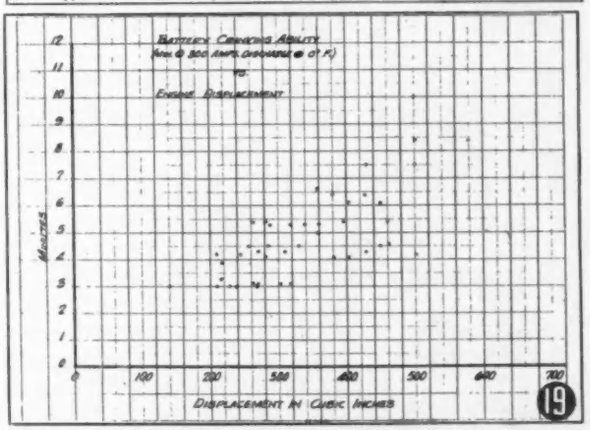
The use of lower viscosity oils in the engine during the winter months also has contributed materially to the solution of the cranking problem. Fig. 20 shows the relationship between engine crankshaft torque and piston displacement for two different oil viscosities.

Referring to this curve, it will be seen that, with a 300-cu.-in. engine 142 ft-lb. torque are required for cranking at 0 deg. fahr. temperature with 20,000-sec. oil. With 45,000-sec. oil 188 ft-lb. are required, or an increase of over 30 per cent which could easily mean the difference between starting









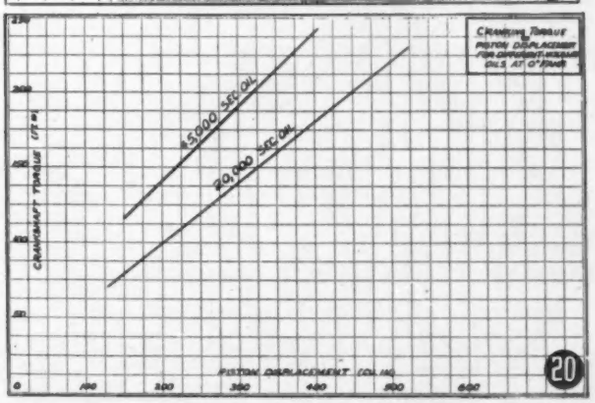


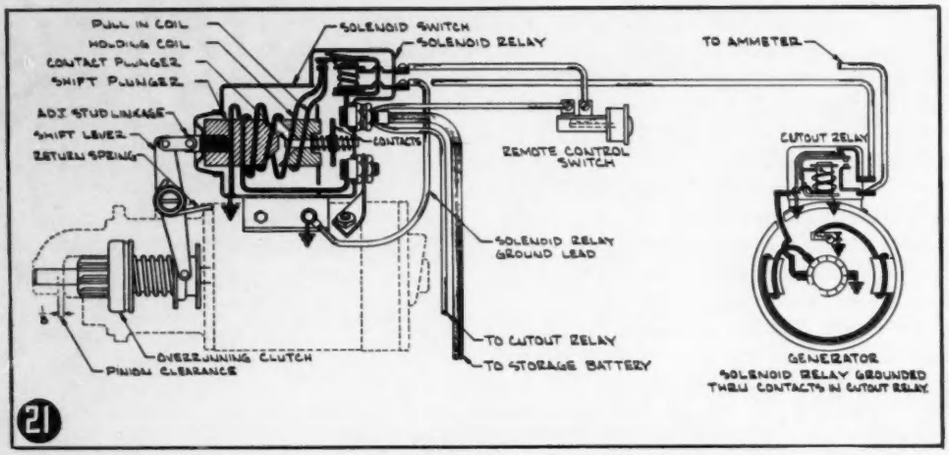
Fig. 16 – Performance Curve of Shunt Generator with Current Regulator

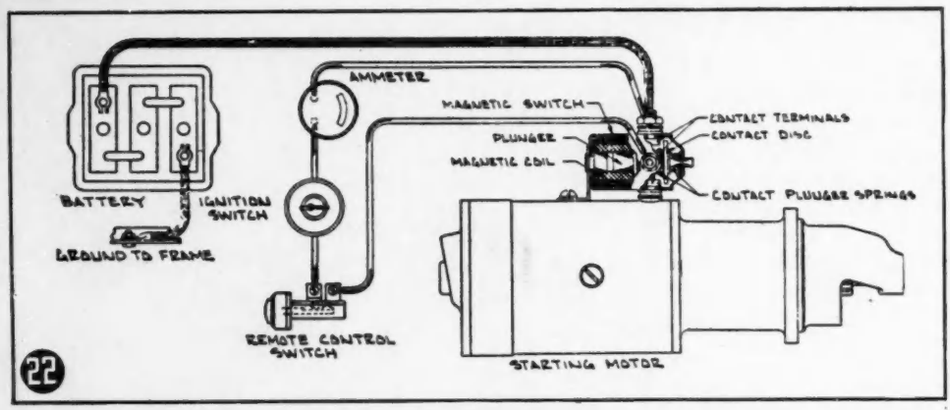
Fig. 17 – Comparison of Hot Outputs of Various Generators

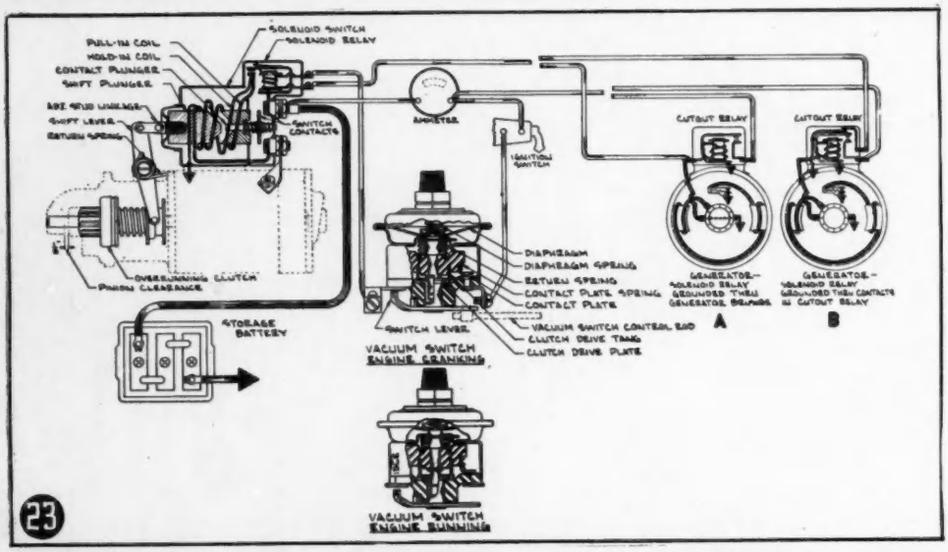
Fig. 18 – Performance Curve of a 1500-Watt Bus Generator

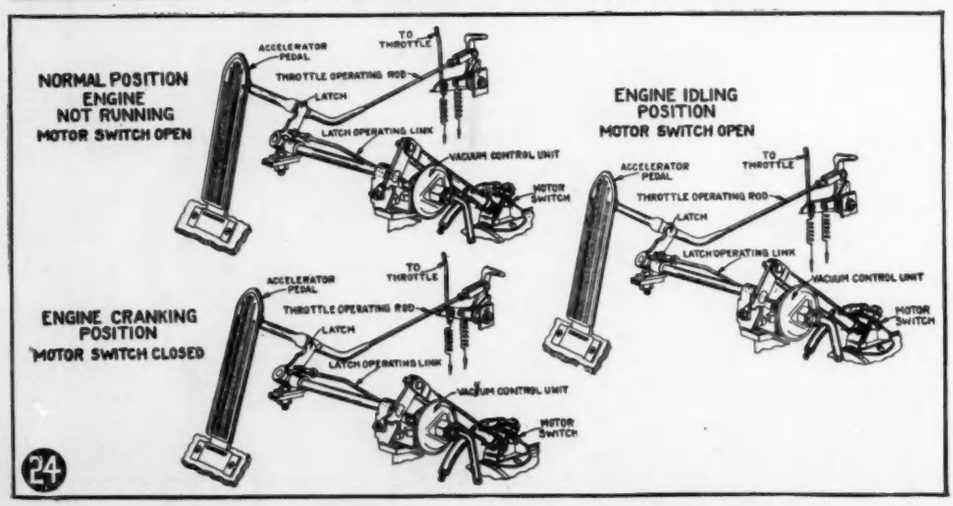
Fig. 19 – Battery Cranking Ability Vs. Engine Displacement

Fig. 20 – Cranking Torque Vs. Piston Displacement for Different Viscosity Oils at 0 Deg. Fahr.









and not starting on a cold morning. Much has been accomplished in the education of the driving public during the last couple of winters as to the correct types of oils for cold-weather use.

Another item in the system which has contributed to the improvement in starting is the means of engagement between the starting motor and the engine. The following pinion engagement systems are in use at the present time:

Overrunning clutch with pedal shift; overrunning clutch with solenoid shift and pushbutton control; overrunning clutch with solenoid shift and accelerator-pedal control; Bendix with foot switch control; Bendix with Starterator control; Bendix with solenoid switch and pushbutton control; Bendix with Startix control; Dyer with pedal shift; Dyer with solenoid shift and pushbutton control. Six of these methods are illustrated in Figs. 21 to 26, inclusive.

This type of drive shown in Fig. 26 has been found to be very satisfactory for heavy-duty service as experienced in bus and Diesel-engine applications. It might be said in passing that, although the overrunning clutch is manually engaged and also manually disengaged, and the Bendix-type drive is automatically engaged and disengaged with the engine ring-gear, the Dyer drive is manually engaged and automatically disengaged.

As stated previously, introduction of Diesel-type engines in the automotive field has increased considerably the starting-motor requirements. A Diesel engine of the same cubic-inch displacement requires considerably more power to start, due in part to the heavier construction and higher friction, in part to the higher compressions used which impose heavier loads on the friction surfaces, but mainly to the fact that engines of this type require cranking at speeds from 125 to 200 r.p.m. while starting, whereas engines of the gasoline type require speeds from 30 to 40 r.p.m.

Fig. 27 shows the horsepower outputs of several starting motors.

The curve down in the lower left-

Fig. 21 – Overrunning Clutch with Pushbutton Control

Fig. 22 – Bendix Drive with Pushbutton Control

Fig. 23 – Overrunning Clutch with Solenoid Shift and Accelerator-Pedal Control

Fig. 24 – Starterator Control Used with Bendix Drive by Chevrolet

hand corner of this figure represents the output performance of a passenger-car starting motor. It will be noted that this motor has a maximum output of approximately 1 hp. The next curve shows the output of what has been considered a large 12-volt starting motor, and the maximum output is 2½ hp. The next curve is for a small Diesel 24-volt starting motor, the maximum output being 7½ hp. The other curve is for a large 24-volt Diesel starting motor. Its maximum output is approximately 12 hp. Not only does the starting motor have to be considerably larger for Diesels, but a larger, higher-voltage battery is required. The extra expense of these units accounts in part for the higher overall cost of the Diesel powerplant.

As brought out previously, 24 volts are necessary in starting some of the larger Diesel engines. Some complete 24-volt systems have been used, but there are certain advantages in keeping the system, outside of the starter, 12-volt. This arrangement permits the use of 12-volt lamp bulbs and accessories, which are more reliable, less expensive, and more apt to be available for service. A solenoid switch has been developed (see Fig. 28) which, when energized in starting, connects two 12-volt batteries in series so that 24 volts are available for the starting motor. When the engine is running, this switch connects the battery in parallel and a 12-volt regulated generator is used in a conventional manner.

Ignition

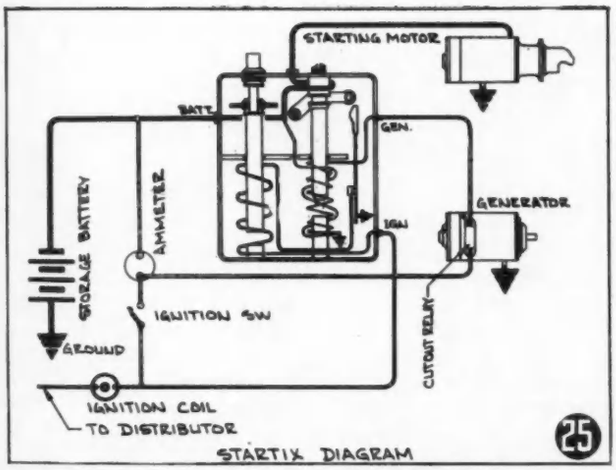
The ignition system consists of distributor timer, coil, high-tension wiring, and spark-plugs, together with a source of electrical energy. No fundamental changes have been made in this system during the past fifteen or twenty years.

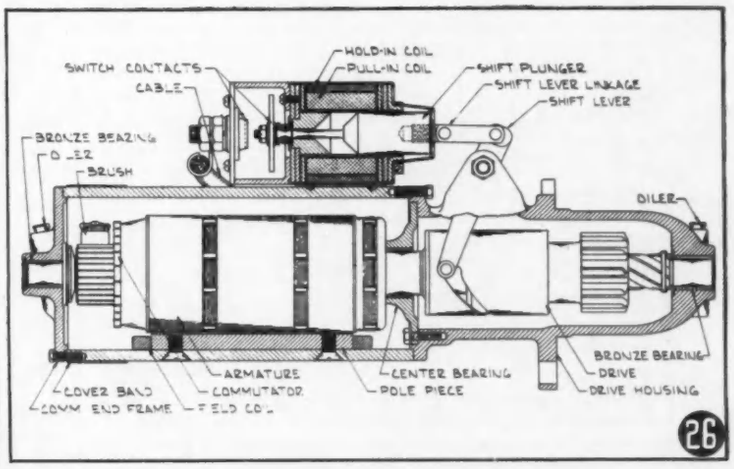
The distributor really serves two purposes. One of these purposes is the proper timing of the opening and closing of the contacts in the ignition coil primary circuit. The other, as implied by its name, is to distribute to the spark-plug in the proper cylinder the high-tension potential from the ignition coil.

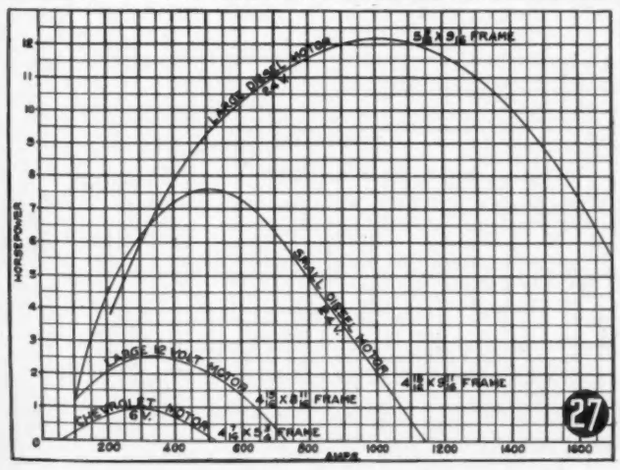
Much has been accomplished in recent years in refinements of distributor design, which improvements have been largely due to increasing knowledge as to the engine requirements and also to the application of common physical laws.

The contacts in the primary circuit must open and close and remain closed for an interval sufficient for the primary current to build up against the coil impedance. In the case of an eight-cylinder engine this cycle of operation must be repeated 12,000 times per mile, or 1,200,000 times in a 100-mile drive. We normally expect to get at least 10,000 miles before contact-point adjustment is required, or a total of 120,000,000 of these cycles. A contact should have a life of 25,000 miles, or 300,000,000 cycles of operation.

The breaker lever must follow the cam without bounce or chatter, and the rate of point opening must be high, so that the primary circuit will be broken cleanly and rapidly and thereby obtain proper rise in the secondary potential. At 80 m.p.h., an eight-cylinder circuit-breaker lever is actuated 16,000 times per min., or 266 times per sec. Of the total time allowable for each cycle of opening and closing, the lever can be raised and lowered in one-third of this total. This operation leaves two-thirds of the time allowable for the contacts







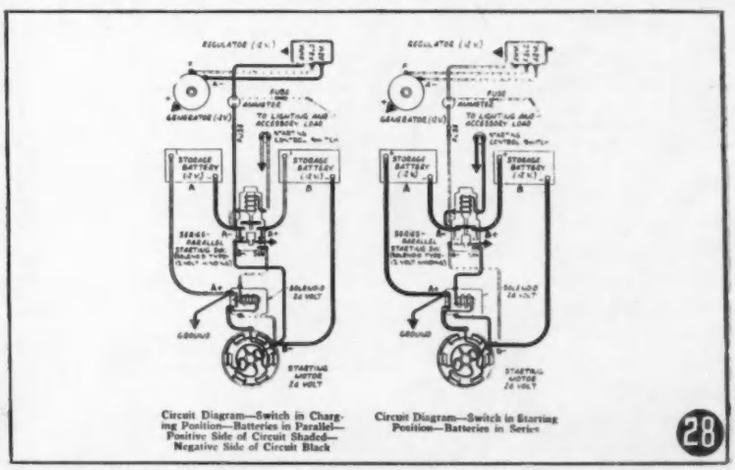
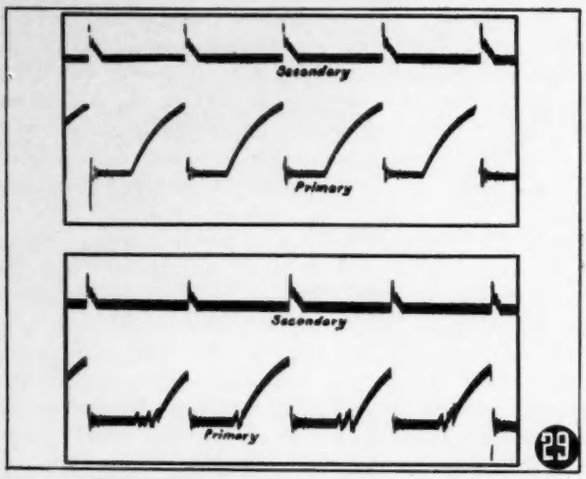

Circuit Diagram—Switch in Charging Position—Batteries in Parallel—Positive Side of Circuit Shaded—Negative Side of Circuit Black
Circuit Diagram—Switch in Starting Position—Batteries in Series
28

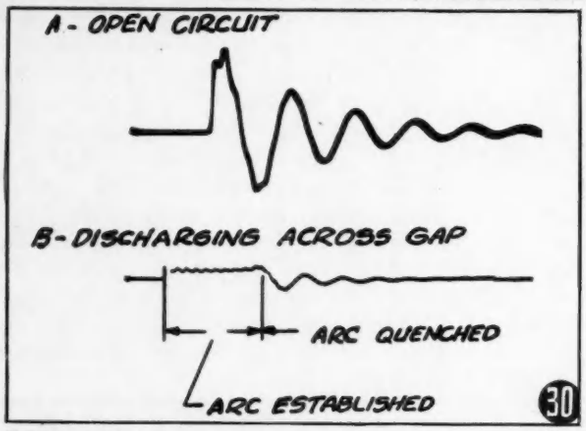
Fig. 25 – Startix System

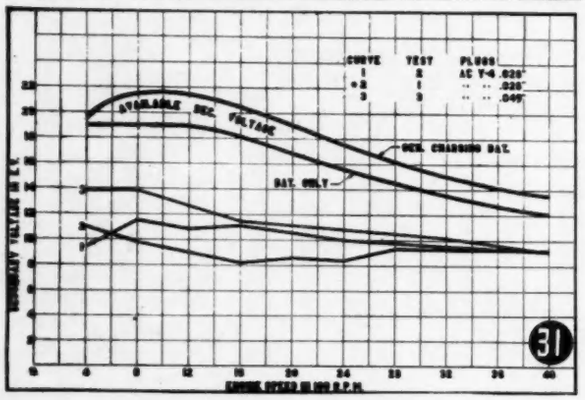
Fig. 26 – Starting Motor Equipped with the Dyer Drive Arranged for Solenoid Shift and Remote Control

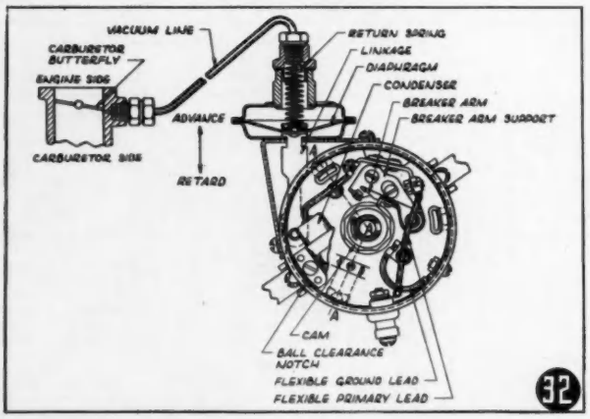
Fig. 27 – Starting Motor Horsepower Output

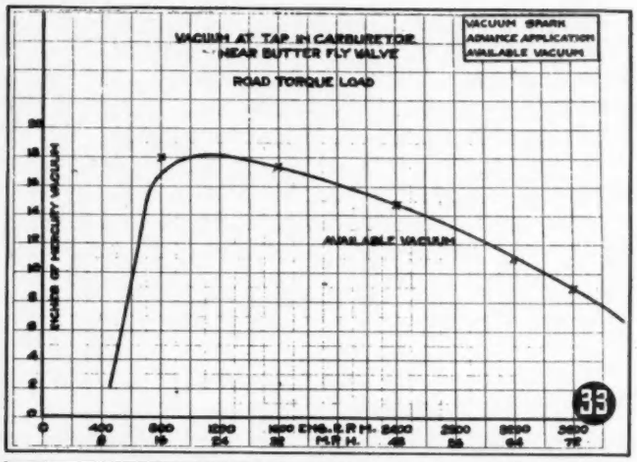
Fig. 28 – Solenoid-Type Series-Parallel Switch

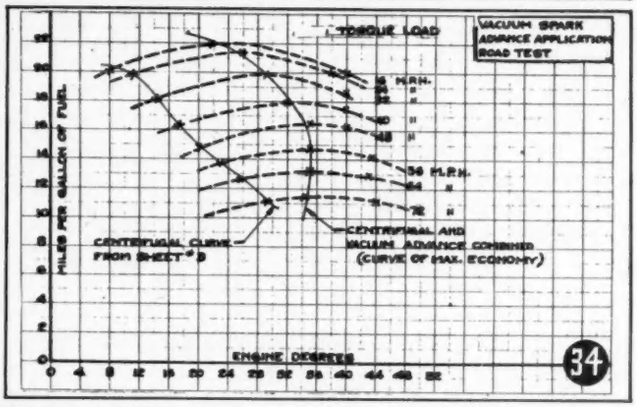


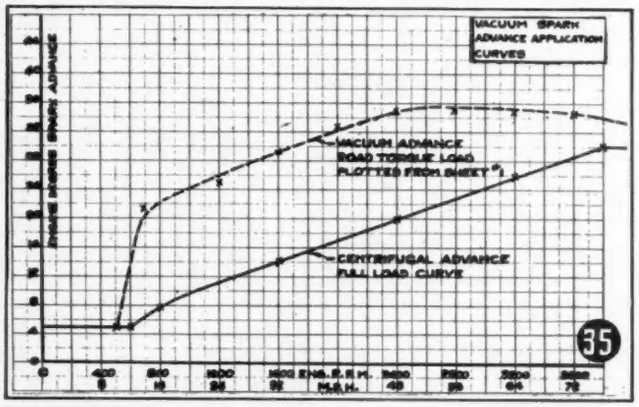









TORQUE LOAD
VACUUM SPARK ADVANCE APPLICATION ROAD TEST
MILES PER GALLON OF FUEL
CENTRIFUGAL CURVE FROM SHEET "B"
CENTRIFUGAL AND VACUUM ADVANCE COMBINED (CURVE OF MAX. ECONOMY)
ENGINE DEGREES
34


VACUUM SPARK ADVANCE APPLICATION CURVES
ENGINE DEGREES SPARK ADVANCE
VACUUM ADVANCE ROAD TORQUE LOAD PLOTTED FROM SHEET "I"
CENTRIFUGAL ADVANCE FULL LOAD CURVE
35

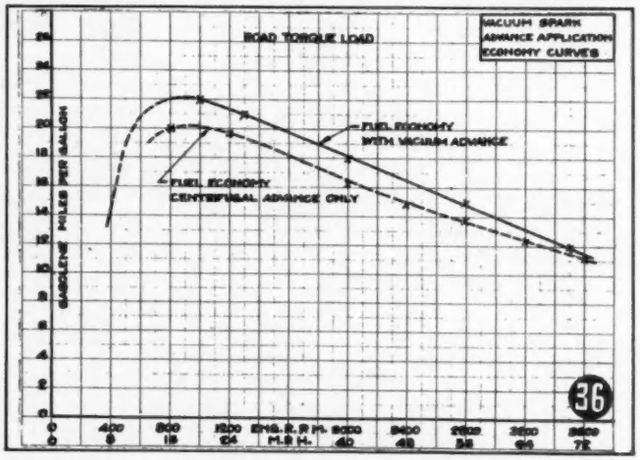


Fig. 29 – Oscillographs Showing the Loss in Ignition Performance Due to Improper Breaker-Lever Action

Fig. 30 – Character of the Secondary Potential

Fig. 31 – Test on Eight-Cylinder 1937 Engine – Curves of Average of Maximum Required Secondary Voltages

Fig. 32 – Vacuum-Spark-Advance System

Fig. 33 – Diaphragm Vacuum Values – Vacuum-Spark-Advance Application

Fig. 34 – Fuel-Consumption Curves

Fig. 35 – Vacuum Spark Advance Superimposed on the Centrifugal Spark-Advance Curve

Fig. 36 – Economy With and Without Vacuum Advance

to remain closed or, under the foregoing conditions, this interval amounts to 0.0025 sec. During this very short interval the current in the primary winding of the ignition coil must build up to an appreciable value. It is obvious that any bouncing or chattering or deflection of the lever will reduce the time of current build-up and hence the energy which can be stored in the windings. This loss, in turn, reduces the energy available for the secondary spark. Fig. 29 illustrates the loss in performance due to improper lever action.

The upper oscillograph on this chart shows the character of the primary and secondary winding current waves with the engine operating at 3500 r.p.m. using a modern design breaker lever. The lower one gives the results of tests on the same equipment but using an old-style breaker mechanism. The effect of the chatter on the lever is very evident, as also is the reduction in primary and secondary peak currents.

Today, maximum engine speeds are in the neighborhood of 4300 r.p.m., and we do not feel that ignition is the limiting factor. Ten years ago 3000 r.p.m. was considered high-speed automobile-engine operation. Twenty years ago 2000 r.p.m. was practically the limit.

As stated before, the opening of the primary ignition circuit by the action of the breaker-lever mechanism causes a flux collapse in the coil, resulting in high potential being generated in the secondary winding, which establishes a spark at the plugs and thus, at the time of opening, the circuit breaker contacts time the spark in the cylinder at the proper time for maximum engine performance. Here accuracy is quite a factor, especially with the modern high-compression engines. A spark slightly early will produce detonation, whereas one slightly late will produce loss in performance. It is customary at the present time to hold the cams to sufficient accuracy that the synchronism of breaker-lever operation from one cam load to another will not vary more than 1 deg. at the cam, or 2 deg. on the engine.

It is interesting to note that some improvement has resulted in ignition performance from the recent trend toward generators of more adequate capacity. With insufficient generator capacity the system voltage, of course, drops to a lower value which reduction, in turn, results in less energy being available in the ignition circuit. At higher engine speeds the better voltage conditions now existing result in noticeably improved ignition characteristics, as will be noted in a later chart.

Instruments which have been made available for study of the transient phenomena of the ignition circuit probably have contributed more to the advance in the present art than any new theory or distinctive development. The cathode-ray oscilloscope and the neon-tube stroboscope have been of practical value in this respect.

Fig. 30 illustrates the character of the secondary potential. Graph *A* represents open-circuit conditions. A peak voltage builds up to between 20,000 and 25,000 volts during the first oscillation, and gradually diminishes in a typical manner at a frequency of approximately 2500 cycles per sec. Graph *B* represents the conditions obtained in the same system except that the secondary gap has been reduced and an arc established for a period of time until the potential is reduced to the point that the arc is quenched, after which the potential wave further diminishes in the characteristic open-circuit manner. It will be noted that, whereas the oscillatory frequency is approximately 2500 per sec. on open circuit, the frequency during time of discharge is approximately 10,000 per sec.

Fig. 31 shows the relation between the voltage necessary to break down spark-plug gap and engine speed for two different values of gap and with two different engine-load conditions at one gap setting. Curves 1 and 3 on this chart give the comparison between two different length gaps, other factors being the same. As would be expected, the lower gap requires slightly less voltage. Curves 1 and 2 are for the same gap, Curve 1 giving the voltage with an engine operating at maximum output, Curve 2 for the engine operating at road torque output. Due to the fact that road torque requires only partial throttle opening and, therefore, less charge is taken into the cylinder, the engine compression is lower, hence lower voltage is required to break down the gap. The upper curves on this chart represent the available secondary voltage, in one case with generator charging the battery, hence with a higher voltage at the coil primary, and the other with the ignition operating off the battery only. The relative heights of these curves with respect to the curves giving voltages required at the plug gaps, indicate the factor of safety for this particular engine. As mentioned earlier in this paper, the benefit of higher voltages obtained with higher output generators is brought out by comparing the available secondary voltage curve with generator charging the battery to that when running off the battery only when the system voltage naturally would be lower.

The distributor, in timing the ignition of the explosive mixture in the engine cylinder and in distributing the secondary potential to the proper cylinder, in most cases also serves another function, namely, it provides automatic advance of the ignition spark with increase in speed. The advance in the ignition spark timing as the engine speed increases is due in part to the time interval required for the propagation of the flame in the combustible mixture which represents increased values in engine degrees as the speed goes up. Other factors influencing the advance required with speed are: plug location in the cylinder; compression ratio; mixture ratio; manifolding and distribution of fuel; valve timing and cam design; octane rating of fuel used; and so on.

The proper timing is obtained by making a full-load dynamometer test on the engine in question at each speed, and the best spark-advance position is determined for maximum power without noticeable spark ping. The curve obtained is generally checked on road test to get maximum acceleration, hill-climbing ability, and so on.

Another function added to the distributor's duties in recent years in the interest of improved fuel economy is that of so-called vacuum control of the ignition spark advance. As stated previously, the automatic advance curve is obtained for the best *maximum* load performance. It is obvious that more advance could be used under part-throttle conditions without incurring objectionable spark ping. This condition is due to the fact that operation under part throttle results in less charge being taken into the cylinder, lower compression and, hence, slower rate of flame propagation. Most passenger-car operation is under part-throttle conditions, so that any gain affecting fuel economy that could be made by altering the spark timing under these conditions is, of course, desirable. Under any given speed condition the manifold vacuum is, in general, responsive to the decrease of throttle opening.

In order to provide this additional control for the ignition timing, a vacuum-operated diaphragm is connected by a tube to the manifold as shown in Fig. 32. This diaphragm is then connected mechanically to the breaker mechanism plate so as to rotate it or, in some instances, it is arranged so as to rotate the complete distributor. Thus, variations in vacuum in the manifold are translated into variations in the timing of the ignition spark (the higher the vacuum, the further advanced the ignition). This variation is superimposed on the automatic ignition spark advance. Maximum manifold vacuum is, of course, obtained under closed-throttle conditions and, during engine idling, the throttle is in this position. Most engines idle better with the spark retarded somewhat. The connection to the manifold vacuum is generally made adjacent

to the throttle butterfly in such a manner as to close off this outlet from the manifold or connect it to atmosphere when the throttle is in the closed position, thus retarding the spark. In obtaining the information desired in making a vacuum-advance application to a particular car, the car is operated at various speeds, and the throttle position noted that is required to maintain the car speed constant under each condition. These throttle positions are designated as those for road load torque. The manifold vacuum is also obtained for each step.

Fig. 33 gives the diaphragm vacuum values for road load torque. This curve illustrates the drop in vacuum obtained at low speeds by proper location of the vacuum tap in the carburetor with respect to the throttle butterfly valve.

The motion of the vacuum-actuated diaphragm, which rotates the breaker mechanism, is restrained by a spring. The proper advance values are obtained by calibrating the size of the diaphragm and the spring, together with the proper location of stops to limit the degrees advance obtained.

The engine is then placed on a dynamometer, and a series of fuel-consumption curves are obtained for each speed and torque condition with different spark-advance values, and the value for best fuel consumption is determined. Fig. 34 illustrates the results of such a series of tests for a particular engine.

Fig. 35 shows the vacuum advance superimposed on the centrifugal advance curve.

Fig. 36 shows the fuel economy with and without vacuum advance for this particular application.

Horns

Automobile warning signals probably have been subjected to more changes in type and design in the past ten or fifteen years than any other unit on the automobile, except possibly the body. This change probably has been due not so much to the fact that the original horns were so bad, either in tone or effectiveness, as to the very definite opinion that each car owner has as to the performance of the horn in his car, and as to how he feels the horn should sound. While on the subject of car-owner opinion on this matter, proof as to the influence of the sound of the horn rather than its effectiveness exists in the average owner's feeling as to the relative merits of the under-the-hood and out-in-front type horn mountings. The out-in-front horn mounting is, in general, more effective as a road signal, whereas the under-the-hood mounting sounds louder to the driver of the car in which it is installed. Therefore, most owners prefer installations under the hood, as they apparently feel that since it sounds louder to them, it must be the better signal.

It is believed that the original horn was of the air type wherein the air pressure was provided by a hand-operated bulb, and the sound-wave-generating unit consisted of a reed and air column similar to that used in a clarinet, or a toy horn.

The next step was to a diaphragm type wherein a manually actuated plunger operated through gearing to rotate a cam against the diaphragm. Both of these horns had the objection that they were manually operated, and the hand-operated-diaphragm horn used by taxicabs and trucks for many years had a particularly raucous sound. Electric-motor-operated diaphragm horns soon followed and remained in production for a longer period of time than any other type up to the present date. A diaphragm type wherein the diaphragm was vibrated by means of an electromagnet was introduced in the early twenties, but was not a very successful design either from the effectiveness standpoint or from the type of note emitted. This type of horn was modified later by the provision of a more powerful electromagnet, and the addition on the front of the diaphragm of a disc which resonated as a harmonic of the diaphragm note, and provided a much more satisfactory warning signal from all standpoints.

Laminated brackets, which permit the horn to vibrate more freely and prevent the horn vibrations from being transmitted to the chassis, were introduced at about the same time and contributed to the overall improvement. Vibrator horns of the resonator-disc type are still in use. Horns operated by difference in pressure, either of steam, compressed air, or vacuum type, have been in use for some time. Due to the manner in which the tone is produced, it has been found necessary to release a tremendous amount of energy in order to obtain effectiveness through the prevailing sound levels and frequencies encountered while driving. Horns of this type, therefore, have been legislated against to a certain extent because of the extreme loudness of the note in their immediate vicinity. The so-called electric-air-tone type of horn has been the latest member of the family, and appears to be the most popular. The characteristic of its tone is such that a fairly effective and pleasing warning signal is obtained without extreme loudness.

Fig. 37 illustrates the characteristic sound waves of representative types of horns.

Examination of the top graph, taken of a motor-driven horn, indicates that, although there is a certain uniform repetition to the fundamental note, there are also waves which are random in form, but continuously maintained. The diaphragm in this type of horn is set in motion by a rotating cam striking a hardened anvil mounted in or near the center. While providing a displacement along the diaphragm axis, there is also a force component at right angles which, together with inherent irregularities in the production cams, accounts for the rough, grating type of tone emitted by these horns and indicated in the sound wave.

The sound wave of the tone emitted by the vibrator horn equipped with resonator disc indicates a more uniform wave structure composed of a fundamental and certain continuously maintained harmonics. This horn gives a more pleasing overall tone, although it contains certain high-pitched frequencies which are displeasing to the ear.

The lower wave graph, representative of the electric-air-type horns, is similar to sound waves obtained with wind-operated instruments such as cornets, trombones, and so on. The tone still carries a certain amount of mechanical noise. The resulting timbre of the note differs slightly from that of a musical instrument. However, the sound emitted combines effectiveness together with the most pleasing warning signal yet introduced. The frequencies of horns of the air-tone type, which are generally used in pairs, are measured on special meters which have been developd and can be held to a tolerance of ± 3 per cent.

Quite a few articles have appeared recently on the subject

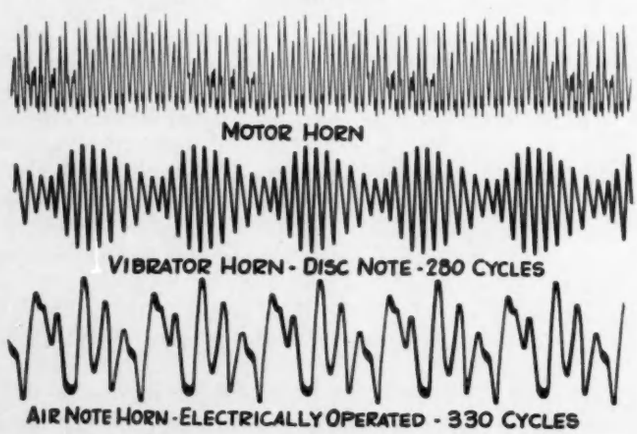


Fig. 37 – Characteristic Sound Waves of Representative Types of Horns

of city noise and the resulting nuisances arising from various noise-producing units, among them the horn. Following are listed the decibels intensity of various common sounds, together with those of representative types of horns.

Ordinary conversation	60 decibels
Inside passenger car (20 m.p.h.)	66 decibels
Inside passenger car (60 m.p.h.)	82 decibels
Orchestra	80 decibels
Pneumatic riveter	101 decibels
Threshold of pain in ear	130 decibels
—	
Bulb horn	76 decibels
Motor horn	91 decibels
Vibrator horn	94 decibels
Pair of air-tone horns	96 decibels

Our experience indicates that the sound intensity is not necessarily a measure of the effectiveness of the horn as a warning signal, as the human ear is more responsive to certain frequencies and combinations of frequencies than to others, and there are sounds of certain frequencies prevalent in the average automobile which require that the horn frequency, in order not to be blanketed, must fall within certain bands.

Horn manufacturers are being pressed on one side for more effective horns, especially for high-speed driving, while at the same time certain noise-abatement committees and commissions are clamoring for reduction in the horn sound level, especially in the larger cities. A high-frequency horn, having slightly different overtone structure, and adapted for outside mounting, has been developed recently to provide increased effectiveness at high speeds. This horn, although having practically double the effectiveness at 60 m.p.h. of the under-the-hood mounted type, has a very disagreeable sound when operated alone, and is, therefore, generally operated as an accessory in conjunction with the equipment horns. A speed responsive switch has been developed which is used to connect this horn in the circuit at higher driving speeds, when it is assumed the car is being operated in the country. This arrangement prevents the horn from being operated at speeds customary for city driving.

In an effort to mitigate the automobile-horn noise evil for city driving, there are under development means for reducing the volume of the equipment horns and making this volume subject to control according to the speed at which the car is being operated. It could thus be possible to have a combination whereby a signal having an intensity of around 10 per cent of that of the present equipment horns would be available for city driving, and the intensity of the present horns could be made available at speeds above 35 m.p.h., while the high-frequency horn note could be superimposed at speeds over 60 m.p.h. to take care of high-speed driving.

Conclusion

The preceding presentation has of necessity been rather general, and has only touched on the general requirements and development of the various units.

It is believed, however, that the increase in number and complexity of electrical units and their importance in the operation of the modern car, are apparent.

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(*Continued from page* 357)

than in the past. These high ratios are inconvenient at low speeds in cities where it is necessary to make many sharp turns. The high ratios are not objectionable at moderate and high speeds, where the wheel does not turn through large angles even when turning in and out of a driving lane passing other traffic. However, when the tire blows out, or a skid develops, a car may take an extreme angle to the necessary path, and any undue delay in adjusting the front wheels to the conditions may be disastrous. Steering ratios are so high that the really skilled driver has difficulty in dealing with the above situation as effectively as possible. It must also be remembered that the angle through which the wheel must be turned includes first the angle required to aim the front wheels at the desired path, and next a sufficient angle to develop the side force needed to straighten out the car. Any sluggishness in responding to the new steering angle will increase the angle through which the steering wheel must be turned and increase the trouble.

Many good observers have the opinion that a faster steering gear would be a source of trouble in the hands of the average driver, and that very fast ratios would be likely to be worse than slower ones even when a tire blows out. This point requires study, but it may well be doubted if the overall effect of increasing the steering-gear ratio has been to reduce the accident rate.

The discussion above which is obviously presented from the viewpoint of the passenger car only, could not have been written without benefit of the advice received from other engineers, and from several persons who are making a special study of the highway safety problem. Information obtained from the writer's associates in General Motors has been invaluable. In particular, acknowledgement must be made to Maurice Olley for a theory of car handling, which is only touched upon in the section on "Roadability." None of these persons should be held responsible for any of the opinions expressed as to the desirability or otherwise, of any particular structural arrangement or detail, as the writer had made all decisions as to points to be mentioned, and the emphasis to be given individual points.

Public officials and engineers interested in safety may feel that too much of this discussion is negative, in that the disadvantages of various proposals are more frequently presented than the advantages, and that too few definite proposals for change are made. This in part may be due to faults in the writer's viewpoint, but it is at least partly due to the method the engineer must follow in attacking any problem upon which there is not enough authentic data to justify a definite conclusion. The engineer must outline all possible objections to an added device on a motor car, whether that device be expected to increase purchaser demand, or to promote safety, and then seek to avoid these objections as far as may be possible in the details of design.

When an automobile engineer asks why it is believed that a certain device would reduce accidents, he is not trying to oppose progress, he is simply trying to find out what proved facts may be available.

The car engineer has learned that while he can frequently design a car which in his own hands will meet certain prescribed standards of performance, he is never sure just what the public will do with this car, and what quite unexpected treatment will be given to it.

Open-minded discussion among the car engineers, public officials, and safety engineers is necessary if any real progress is to be made. By this method, we can hope to reduce the time and study necessary to determine what changes may be really useful in protecting our drivers against themselves, and against each other.

It is hoped that the above incomplete discussion will stimulate more adequate discussion of the topics mentioned, and of others, on the part of persons equipped to bring forward facts where the writer could only suggest opinions.

The "Tocco" Process of Hardening

By W. E. Benninghoff

The Ohio Crankshaft Co.

THE "Tocco" process of surface hardening, especially of the bearing surfaces of bus and truck crankshafts, is described in this paper. Desirable changes and modifications in production operations and their sequence made possible by the process are suggested. Examples are reported from service experience of the amount of resistance to wear and abrasion added to the bearing surfaces by this process.

THE growth of intercity bus and truck service is in a way a measure of the development of the heavy-duty engine and especially of the crankshaft. The earlier engines were equipped with heat-treated carbon-steel crankshafts, but serious difficulty was experienced with them due to bearing wear as the size of loads and length of runs increased.

In order to obtain harder, more wear-resistant bearing surfaces, the engine designers then resorted to alloy steels, heat-treating them up to the limit of machinability. These shafts, of course, showed a marked improvement in bearing wear. As motor development progressed and with the advent of the automotive-type Diesel engine and of harder bearing materials, there was insistent demand for still better wear-resistant bearing surfaces. In other words, the motor designer, the truck and bus salesman, and the fleet owners demanded a crankshaft that would be capable of operating many thousands of miles without a bearing failure or even a bearing adjustment.

Carburizing, nitriding, and many other methods of surface hardening were tried to produce the desired results. All of these methods were found wanting in one particular or another. A method was needed that would produce the desired surface hardness with a structure which would withstand wear and abrasion without being subject to flaking or spalling. It must be accurate as to control of the depth, width, and location of the hardened area and give absolute uniformity of hardening from bearing to bearing and crankshaft to crankshaft. It must have a short time cycle in order to meet production requirements, and the method must not be expensive.

With these somewhat high marks to shoot at, the Tocco process was perfected after a period of research and development work. The name, "Tocco," incidentally, is a coined word formed by combining the first letters of the company name.

In brief, the process consists of the following:

High-frequency current of 2000 cycles is passed through an inductor block which surrounds but does not touch the bearing to be hardened. This high-frequency current produces a very strong magnetic field which cuts the bearing surface through a small air gap, and induces eddy currents in the bearing surface. The magnetic field also creates hysteresis losses in the bearing surface. The combined eddy current and hysteresis losses, therefore, cause heat to be generated in the surface of the steel itself. The inductor blocks remain cold.

Due to the inherent reaction of the steel as its temperature rises, the heating effect decreases as the critical point of the steel is approached and, consequently, the surface of the steel is not overheated since continued application of power would only cause the heat to be generated farther in toward the center of the shaft. The mathematics of this heating is quite involved since it is dependent upon the strength of the magnetic field, the frequency of the current, the width of the air gap between the block and the shaft, the steel analysis, the grain size of the steel, the temperature of the steel, as well as the electrical resistance and permeability of the steel. The electrical theory of the Tocco process would furnish sufficient material for several discussions.

With the application of high-frequency current to the inductor block, the bearing surface is heated to the quenching temperature for a depth of approximately ⅛ in. in approximately 5 sec., and is then immediately quenched by pressure-spraying through orifices drilled through the inductor blocks and into a water jacket which is integral with the block.

The rate of power input and length of time that it is applied vary with different sizes of bearings as well as with different steels. The kilowatt-seconds required for a particular bearing are calculated from graphs compiled from previous data, with due allowance being made for the mass effect of adjacent areas such as crank cheeks. This calculation is then checked by experimentation.

Due to the speed of the cycle which is measured in split seconds, there is no scaling or pitting of the bearing surfaces, which remain as clean as they were before surface hardening. There is, however, a change in color. Furthermore, since only the bearing surfaces are heated, there is little chance for distortion of the shaft.

Automatic Operation

All the control factors, such as rate of power input, length of heating cycle, quenching pressure, and time of quenching are automatic. The operator's only function is to place the shaft in position and push a button to start the Tocco cycle. Once the control has been set to produce a certain result on a certain piece, all the pieces will be absolute duplicates in the heat-treatment given them.

The shafts which are to be Tocco hardened usually are forged of a modified S.A.E. 1045 or 1050 steel and either heat-treated or normalized for maximum ductility consistent with the physical properties desired. The shaft is then machined to the point of final grinding with 0.015 to 0.025 in. of stock left on the bearing diameters and 0.010 to 0.015 in. of stock left on the bearing widths. The bearing surfaces are then hardened, the shaft drawn at a low temperature, and then finish-ground. The bearing hardness obtained is in the range of 58-62 Rock-

[This paper was presented at the Philadelphia Section Meeting of the Society, Philadelphia, Pa., Jan. 20, 1937.]

well C, which is approximately 85 scleroscope or 600 Brinell, and this hardness is maintained through about 80 per cent of the depth of the hardened area.

The structure in the hardened area is extremely fine-grained martensite and, even at 1000X the usual large needles associated with martensite are absent. This structure results in exceptional ductility throughout the hardened area. Furthermore, due to the rapidity of the heating cycle, surface decarburization and grain growth do not occur.

The combination of accurate control, split-second heating cycles, and pressure-quenching applied instantaneously at the end of the heating cycle, produces a bond – that is, the cohesion between the hardened area and the core – which blends gradually into the core structure without any marked precipitation of free ferrite either in or below the gradation zone. The gradual blending of the hardened area into the core without precipitation of ferrite prevents any flaking or spalling in service. The 175,000 to 200,000 shafts now in heavy-duty service have not had a single failure in the bond or the hardened zone.

Original Structures Undisturbed

The original structures in the fillet and the core remain entirely undisturbed since the heating is confined to the areas desired hardened. Since the fillets are not hardened but left in their original ductile condition, stress concentration, therefore, is decreased. Many engineers place considerable emphasis on fatigue values as determined in the fatigue testing machine without giving proper consideration to stress concentration. It is a well-known fact that alloy steels produce better fatigue values than carbon steels in the testing machine, but they also show a greater tendency to stress concentration.

As an illustration of this point, an engine manufacturer was using an alloy crankshaft and, in order to reduce bearing wear, he heat-treated it to the limit of machinability. The shafts began to break after about 30,000 miles. The failures were analyzed as being caused by fatigue due to stress concentration in the fillets. Without changing any factor except the crankshaft steel, which was changed to S.A.E. 1045 steel, with the bearings Tocco hardened for wear resistance, the failures ceased. Incidentally, there are several records of these shafts running 150,000 miles without perceptible bearing wear.

We can now, therefore, approach the ideal crankshaft – one with maximum ductility to withstand physical abuse and with high bearing hardness to withstand wear and abrasion.

As an illustration of what the hard bearing surface with its accompanying lack of bearing wear means to the bus or truck operator, the following statements were made by operators to the engine manufacturers using the surface-hardened shaft:

"The oil-pressure gage reading after 80,000 miles of service had not dropped" . . . "former mileage period before regrinding crankshafts increased four to five times" . . . "100,000 miles without a bearing adjustment" . . . "maintenance of new engine rate of oil throw-off from the crankshaft eliminates increasing carbon and piston-ring troubles" which usually develop as the crankshaft wears out of round or undersize.

In one specific instance, a Diesel engine operating in bad alkali-dust conditions required regrinding of the crankshaft after 750 hr. of service with the ordinary heat-treated crankshaft. The surface-hardened crankshaft put in the same engine was allowed to run 4000 hr. before it was taken out for examination. Measurements showed no appreciable wear, and it was replaced in the same engine.

A further development, where crankshaft design permits, is to machine the forging in the normalized condition instead of the heat-treated condition, Tocco-harden and finish-grind without further heat-treatment. This method presents the obvious advantage of saving in heat-treating costs. The normalized structure is easier to machine, saving approximately 25 per cent of the machining costs and lengthening tool life 50 per cent and, furthermore, it does not require as many or as costly straightening operations during machining. This development permits the use of the hard-surface shaft with little or no increase in cost.

Due to the fact that new combinations of bearing materials with the surface-hardened shaft will permit higher bearing speeds and pressures, the newer engines are being designed to take full advantage of these features with a consequent increase in horsepower per cubic inch of displacement or, stated another way, the weight per horsepower can be decreased with a corresponding effect throughout the vehicle.

Although this discussion has been limited to crankshaft practice, Tocco hardening also can be applied to axle shafts and similar applications, such as wheel spindles with the rollers riding directly on the shaft. This arrangement eliminates the inner roller race, and the space formerly occupied by the race is used to allow more metal to be put in the shaft at that point and thus increase its strength.

In fact, any surface subject to wear can be surface-hardened locally without heating the entire shaft. The long time cycles with the resulting distortion and excessive cost to heat-treat the whole shaft to harden just a small surface are eliminated. In a large crankshaft weighing approximately 1000 lb., only 30 lb. is required hardened for bearing surface.

A more recent development is the hardening of camshafts. Inductive heating now can be used to harden irregular shapes. The camshafts are machined easily in the normalized condition, the cams and bearing surfaces, if necessary, are then hardened, and the shaft finish-ground. The long time cycles of carburizing with the consequent distortion, copper plating, and expensive straightening, are eliminated. Since only the surfaces are heated and the cycle is short, there is little distortion. The same hard structure of 60 Rockwell C is obtained in the hardened area.

To summarize briefly, the high surface hardness is controlled accurately as to depth, width, and location with no change in the adjacent areas. The bond between the hardened area and the core is well blended, and results are duplicated, insuring absolute uniformity. High bearing hardness is obtained without impairing the ductility of the shaft, giving maximum wear and abrasion resistance, as well as maximum resistance to physical abuse.

Servicing Diesel Engines

IT should be stated at the outset that we have never regarded Diesel-engine service as a problem demanding a separate service setup for its solution – and there has been a very good reason for it. The Diesel engine itself, exclusive of the injection system, did not at its introduction present any service problems with which the branch and dealer service organizations were not already qualified to deal by virtue of their experience in servicing more than a million and a quarter conventional engines during a quarter of a century. There was nothing about the Diesel, except the injection system, with which any trained service man was not already familiar in principle. The service man found the same simplicity and accessibility as characterized the conventional models.

As the injection system represented the only important departure from standard design and practice, an exchange pump service plan was put into effect while the organization was being schooled in this ultra-precision-made equipment.

Excerpt from the paper of the same title by R. J. Kretz, presented at the Tractor Meeting of the Society, Peoria, Ill., April 23, 1937.

The Vibration Problem in Aircraft-Propeller Designing

By F. W. Caldwell

Hamilton Standard Propellers, Division of United Aircraft Corp.

THIS paper represents an effort on the part of the propeller designer to look at some phases of the vibration problem as it affects him. A very brief description of some of the work being done in vibration is given. The subject is treated from the aspect of experimentation and the physical phenomena without any effort to introduce the mathematical phase of the subject.

Examples are given of the measurements of the vibratory stress in propeller blades by a new method introduced during the last year.

THE writer's first connection with the propeller vibration problem occurred during 1916 in connection with attempts to relate the ordinary static and elastic stress calculations to the actual stress of propellers on test. At that time we were working almost entirely with wooden propellers and the flexural vibrations were not particularly troublesome. Due to the weakness of the wood in torsional shear along the grain and due to the unstable aerodynamic effects in torsion, the problem of flutter was a rather troublesome one and a good many failures occurred from this cause.

The effect of the centrifugal force in balancing the air-load bending moments was such that calculated stresses did not seem to change very much with thickness of the blades, but we soon learned that a certain minimum thickness was required to avoid failure due to flutter, and we found it necessary to rely more on test than on calculation. As a result, we adopted the practice of carrying out whirling tests at a power input 50 per cent in excess of the power output of the engine on which the propeller was to be used.

This type test of 10 hr. at 50 per cent above the normal power proved to be fairly satisfactory for the wooden propellers and, in general, there were hardly any failures of designs which had been subjected to this type of test. However, even in this early experience with wooden propellers, the vibration problem was troublesome at times in connection with the attachment of the propeller to the steel hub. In cases of very rough engines, we occasionally observed charring of the wood surfaces in contact with the steel hub as a result of the engine vibrations. If allowed to continue, this charring would lead to complete failure of the propeller at the hub.

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 11, 1937.]

In going from the wood propellers to the micarta type, the 50 per cent overload test was found inadequate and was replaced with a 100 per cent overload. This test was fairly satisfactory, although an interesting example of vibration effect not found on the electric-motor test was observed in connection with a micarta propeller built for a 300-hp. eight-cylinder V-type engine. In this case the excitation was so severe that internal working of the material resulted in overheating to such a point as to char the surface and lead to failure of the blades, as shown in Fig. 1.

When we came to the adoption of the metal propellers, it was found that the overload test at twice the normal engine power was insufficient to insure against fatigue failure in actual service and the test requirement was increased to test-

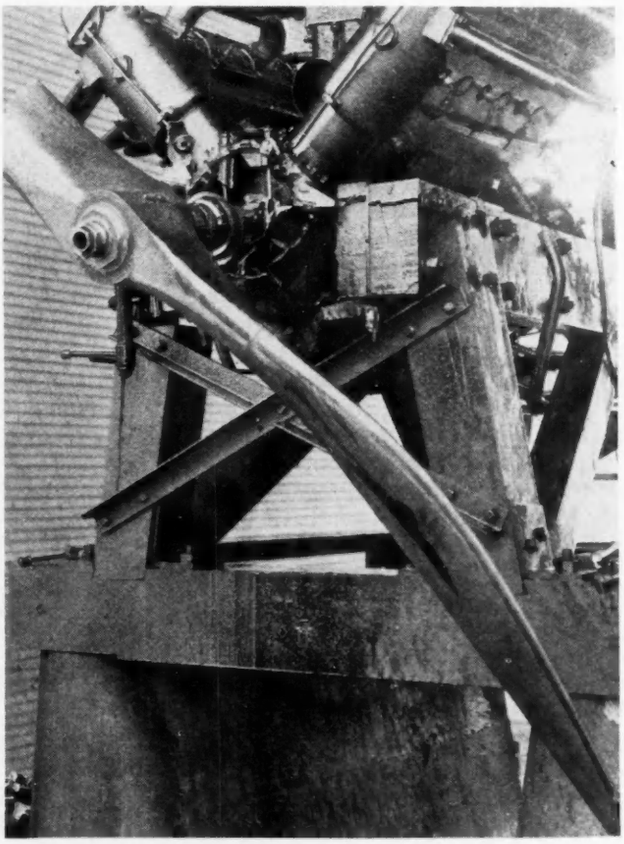


Fig. 1 – Effect of Internal Damping Caused Charring of Micarta when Subjected to Severe Vibratory Stress in Operation

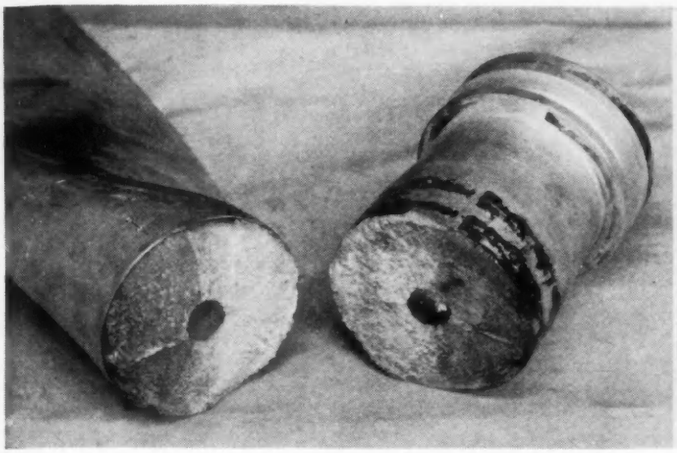


Fig. 2 – Typical Fatigue Failure of Adjustable Blade at the Root Due to Vibration Stress

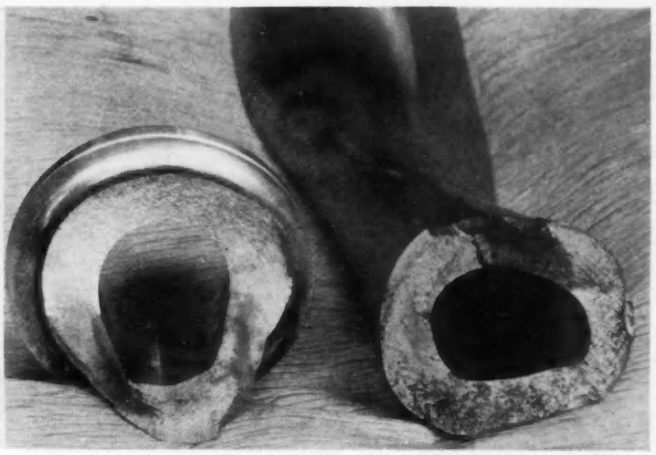


Fig. 3 – Failure of Controllable Blade at Root Due to Violent Crankshaft Resonance – This Is the Only Test Failure so far Experienced on the Controllable Blades – Up to the Present Time None Have Failed at the Root in Service

ing at three times engine power. This test, however, was still insufficient to be a satisfactory criterion, as the vibration stresses were even more predominant in the metal propeller, and the practice of following the whirling test by test on the engine at maximum power output was adopted. This test was set initially at 10 hr. on the test stand but has been progressively increased to a period of 100 hr. It is still considered inadequate from the standpoint of a reasonable factor of safety unless other considerations are taken into account. Propeller failures due to vibration stress and to violent crankshaft resonance are shown in Figs. 2 and 3, respectively.

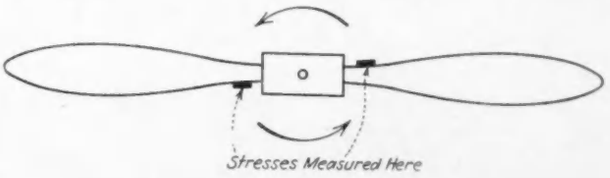
Sources of Excitation

The most common of the studied sources of excitation for propeller vibrations is the rhythmic engine explosions. Since an enormous amount of energy is available for excitation from this source, it is probably the most important single disturbing factor.

However, there are numerous other sources of excitation such as the unbalance occurring at twice crankshaft speed, the crankshaft-propeller-shaft resonance, and torque surges in the engine, the tuning-fork type of vibration in the crankshafts, and the motion of the engine itself in space along and about the three axes. A graphic record of the stresses at the blade roots of a two-bladed propeller caused by torsional resonance of the crankshaft is shown in Fig. 4.

Fig. 4 (*above and at right*) – Example of Stress in Blade Roots Caused by Torsional Resonance of the Crankshaft

This record of stresses at the blade roots of a two-bladed propeller was taken when the crankshaft was operating at torsional resonance excited at a frequency of four-and-one-half times engine speed. The strain gages were located on the leading edges of the two blade shanks.



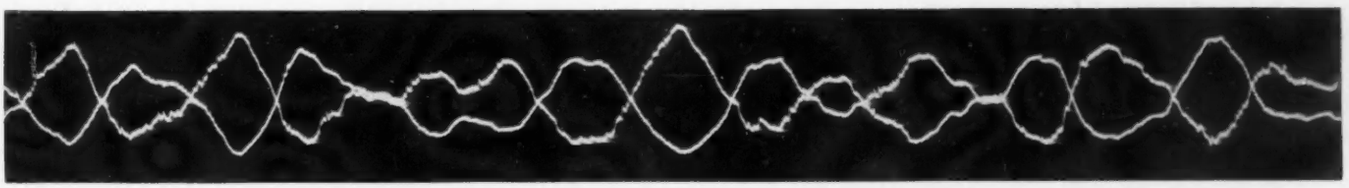


Fig. 5 – Example of Stress Thought to Be a Combination of Torsion and Lateral Motion of Engine

This oscillogram shows two blade-shank stress records taken on a three-bladed propeller mounted on a fourteen-cylinder geared engine. The pickups are located on the leading and trailing edges of one blade shank and thus the individual records are 180 deg. out of phase.

Besides the engine explosions there are various kinds of excitation from aerodynamic sources such as passage of the blades past the wing or adjacent to the body. Fig. 5 gives an example of stress thought to be a combination of torsional and lateral motion of the engine.

A further source of aerodynamic excitation which deserves a great deal more study is the effect of the frequency of the vortices flowing over the propeller section. Since this frequency depends on the angle of incidence and helical velocity,

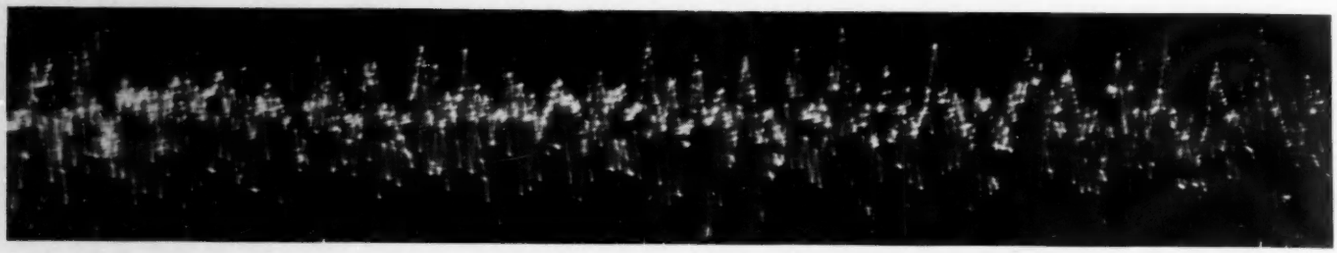


Fig. 6 – Example of Very High Frequency Stress of Great Amplitude (about 40,000 cycles per min.)

This oscillogram shows the stress records of two strain gages located near the tips of a two-way propeller as shown below; one blade is experiencing a large amplitude vibration which is nearly independent of the other at a frequency of 40,000 cycles per min.

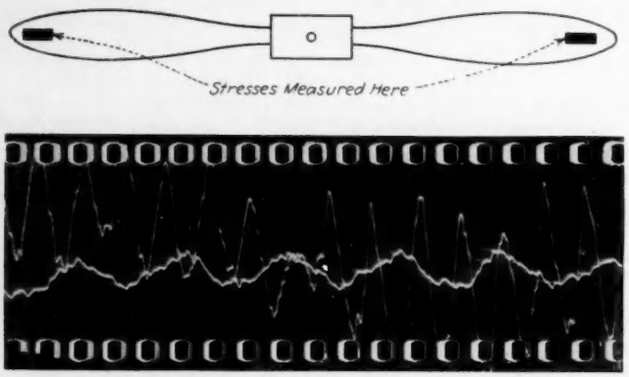


Fig. 7 – Example of High Vibratory Stress Caused by Excessively Rich Mixture Leading to Loss in Power of Lower Cylinders

The torsiograph record shows an amplitude in excess of 2 deg. at a frequency of two and one-half times engine speed. The lower frequency record shows the rotational movement of the engine on the stand at once engine speed. The excitation here is caused by excessively rich mixture leading to bad distribution.

it should vary along the length of the blade and might cause a very wide band of excitation. The frequency is fairly high and may in some cases match with the torsional frequency of the blade, leading to flutter. This theory may explain to some extent the tendency of blades to flutter at higher angles of incidence, since the frequency of the vortex formation is lower as the angle of incidence increases. A record of a very high frequency stress of great amplitude is shown in Fig. 6.

An interesting example of excitation was found during a recent test when the mixture strength of the engine was made excessively rich. The excitation was high enough to cause very high stresses in the propeller blade and the theory was developed that the lower cylinders were enriched to such an extent as to lose a great deal of power, resulting in a rhythmic torque variation at a frequency matching that of the vibration record. This record is shown in Fig. 7.

Fortunately, with the direct-drive engines, the torsional frequency is usually high enough to be outside of the operating range as far as excitation due to engine explosions is concerned. This condition is no longer the case, however, with the increase in number of engine cylinders and crank revolutions per minute and the natural lowering of the blade frequencies, due to the increased diameter required on the geared engines. We probably shall have to watch this form of excitation more closely in the future. The excitation due to torsion of the propeller shaft should be subject to a reasonable amount of control without excessive modifications in our present practice.

The gyrostatic moments of the rotating propeller furnish a source of excitation which in some cases has led to fairly severe propeller vibrations. The gyrostatic couple causes a complete reversal of stress in each blade once during the revolution and, also, a complete reversal of stress in the crankshaft once during each turn. This load, of course, is quite different from the load imposed on the aircraft structure which increases from approximately zero to maximum twice during each turn for two-bladed propellers and which are of fairly steady magnitude for propellers having more than two blades. The calculation of these forces is a fairly familiar subject and need not be mentioned in detail.

Damping

The use of internal damping of materials is probably of considerable interest where resonance vibrations are concerned. An example of this case is the micarta blade where very high internal damping is present.

It is doubtful, however, whether internal damping is beneficial where forced vibration of great magnitude is present, since the work absorbed by the material may lead to destruction more rapidly than would be the case with material of low hysteresis.

Probably the most powerful damping of the flexural vibrations comes from the aerodynamic forces on the blades. A rough calculation will serve to illustrate the possible order of magnitude of these damping forces.

Assuming a fundamental form of vibration at a frequency of 3000 cycles per min. with an amplitude of 3 in. and a

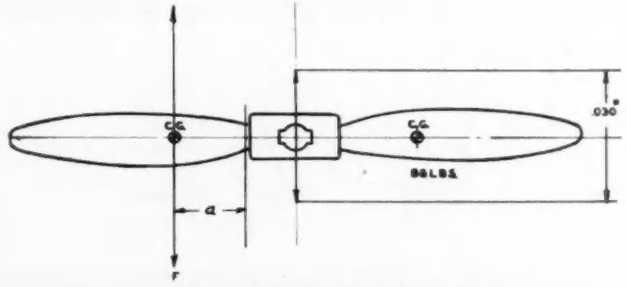

C.G.
C.G.
.030"
88 LBS.
a
F
IF THE PROPELLER WERE OSCILLATED THROUGH .030" DISPLACEMENT AT A FREQUENCY OF 17500 CYCLES PER MINUTE THE FORCE ON A BLADE WEIGHING 88LBS. WOULD REACH A VALUE SUFFICIENT TO CAUSE A STRESS IN THE SHANK OF 25000 LBS PER SQ. IN.

Fig. 8 – Illustration of Forces on Rigid Propeller when Oscillated at Right Angles to its Length

helical velocity (W) of 900 ft. per sec., one sq. ft. of surface would exert an aerodynamic damping effort fluctuating from zero to 13 hp.

Unfortunately, the torsional forces due to aerodynamic loads are unstable so that the damping is negative. The location of the center of pressure ahead of the center of torsional rigidity and the shift of the center of pressure with angle of attack lead to instability. It may be, however, that flexural motion may introduce damping due to coupling between the torsional and flexural modes.

Experimental Investigations

We already have had a great deal of relief from the very severe conditions resulting at shaft resonance speeds with the introduction of the dynamic damper. Although the damper certainly has eliminated the most severe condition, it does not entirely eliminate torsional excitation and it undoubtedly will be very desirable to provide a very much softer drive in addition to the use of the damping device.

Unfortunately, however, the torsional excitation is not the only serious source of stress as the crankshaft may vibrate in a manner similar to a tuning fork giving rise to fore-and-aft displacements, and the whole engine may move along and about the three axes in space.

Some very valuable work is being done at Massachusetts Institute of Technology on the study of the motion of the engine in space, and I am hoping that Prof. C. S. Draper

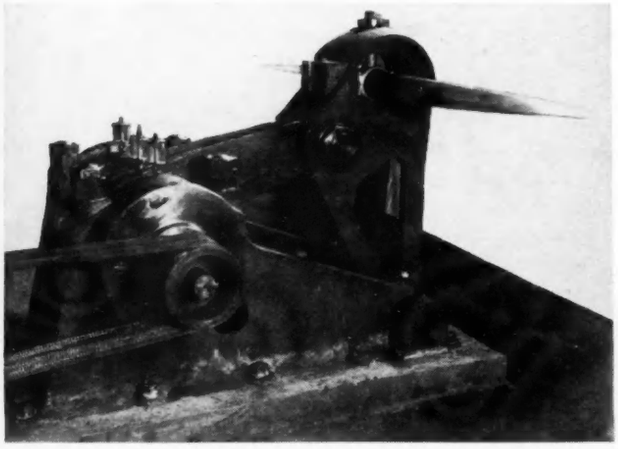


Fig. 9 – Massive Machine for Forced Vibration Tests

of that institution will be able to give us some information on this subject.

We have had a number of indications in our practical experience that serious stresses are introduced from this source

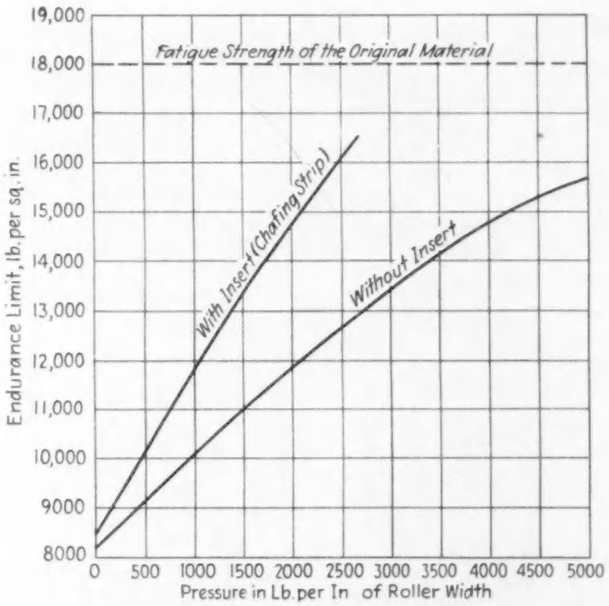


Fig. 10 – Effect of Chafing Strip and Surface Cold Work on the Fatigue Properties of Magnesium Alloys when Subjected to Chafing – Effect on Endurance Limit of Various Rolling Pressures on Elektron Bars when Subjected to Alternating Stress at Point of Attachment to Normal Propeller Hub

The values for the endurance limit are corrected by extrapolation to 10^7 cycles of stress.

Note: The reduced strength below the value of the original material is caused by the chafing action at the point attached to the hub.

and it is interesting to note that a motion of the engine of rather small amplitude at right angles to the length of the propeller blades can produce stresses of a very high order. For example, taking a typical two-blade propeller and treating it as a rigid body, and considering a forced oscillation at right angles to the length of the blade, a motion of 0.030 in. at a frequency of 17,500 cycles per min. would introduce an alternating stress of approximately 25,000 lb. per sq. in. The

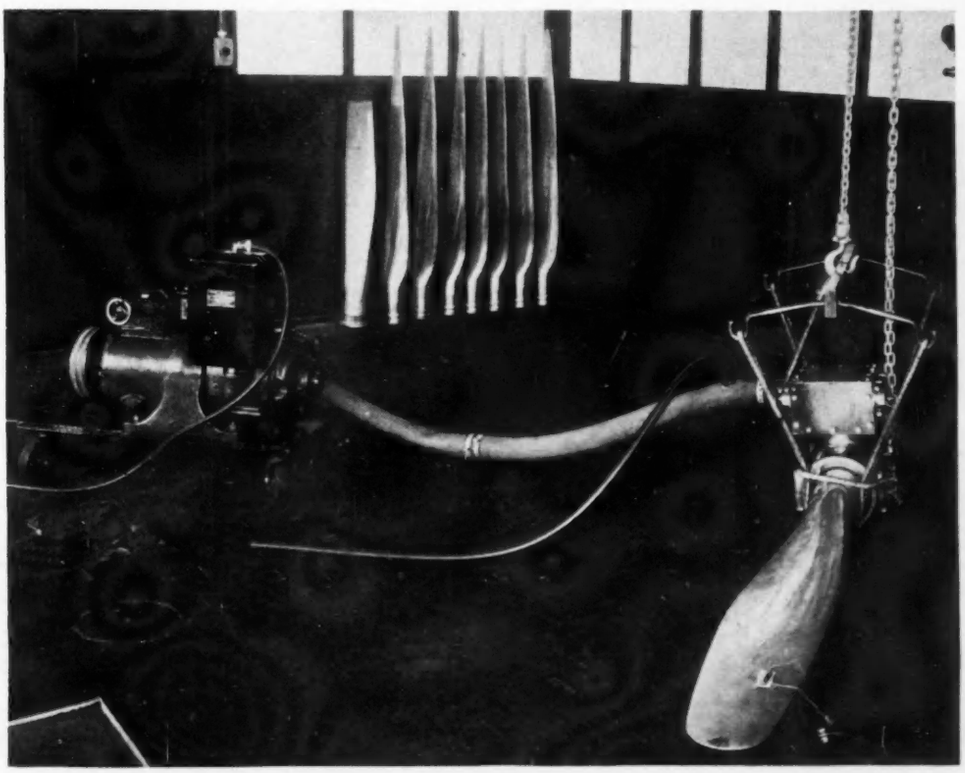


Fig. 11 – Exciter with Linear Oscillator and Transitorq Drive for Speed Control

Fig. 12 – Blade Failure Occurring from Resonance Testing at High Stress – Note Rapid Progression of Fatigue Lines – Second Failure of Forged Hamilton Standard 6106-18 Blade at 25 In. Station after 234 Min. of Vibrating at 17,000 Lb. per Sq. In. and 627,000 Cycles of Stress

forces on a rigid propeller when oscillated at right angles to its length are illustrated in Fig. 8.

With this type of stress in view, our company designed during 1930 a type of machine in which the propellers were subjected to forced oscillation at right angles to their length. Considering the nature of vibration phenomena, a fairly good agreement was obtained between the failures on this machine and those observed in engine test, and a great deal of value was obtained from the tests carried out. This machine is shown in Fig. 9.

It has been necessary to use the forced oscillations in order to produce high stresses near the hub since the various types of resonance vibration tend to produce high stresses in the outer parts of the blades and fail to furnish comparative tests for the hub construction.

An alternate method introduced by the Deutschen Versuchsanstalt für Luftfahrt (D.V.L.) consists of the use of cylindrical bars in place of the propeller blades, these bars being mounted in the usual propeller hub. The bars can be then vibrated in resonance, producing the maximum stresses near the hub, and a comparative study of various constructions can be carried out quite rapidly through this method of test.

Fig. 10 shows the results of some very interesting experiments, carried out by the D.V.L. in this manner, showing comparative strength of the shanks of propeller blades subjected to various degrees of surface cold working prior to installation in the hubs. The tests indicate a very great improvement in resistance to combined fatigue and chafing effects as the result of cold working the material. They also indicate a great improvement by use of a suitable fabric chafing strip between the blades and the hubs[1].

In connection with the resonance test of propeller blades, it has been found possible to produce very high stresses with a very small power input to the source of excitation. For example, with a modern 1-hp. air motor driving an unbalanced flywheel, it has been found possible to produce vibratory stresses above 20,000 lb. per sq. in. in a normal-type propeller blade. Operating over a period of a few hours the blades can be broken in this type of test. See Figs. 11 and 12. The second failure of a forged Hamilton Standard 6106-18 blade at 25-in. station after 234 min. of vibrating at 17,000 lb. per sq. in. and 627,000 cycles of stress, is shown in Fig. 12.

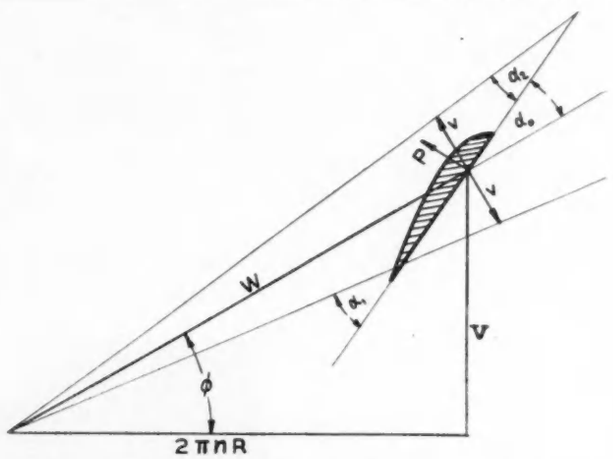


Fig. 14 – Illustration of Effect of Aerodynamic Forces in Damping Flexural Vibrations

When we consider the amount of excitation present in the normal torque variation of the combustion engine, the immediate thought arises that all the propellers should break in a very few seconds near the resonant speed. In Fig. 13 is illustrated an example of a curve of torque due to gas forces for a typical two-row fourteen-cylinder engine of the modern design. Considering the energy of the power loops above and below the mean and taking only the variations occurring at the rate of seven per revolution, the amount of power available for excitation would be 120 hp. There is a sufficiently large amount of power to break the blades in an extremely short period of time, and it is probable that the blades are only saved from failure by the presence of aerodynamic damping of a fairly high order.

[1] See Einige Entwicklungsfragen auf dem Gebiete der Leichtmetall-Walzlegierungen, von Georg Sachs, 1936.

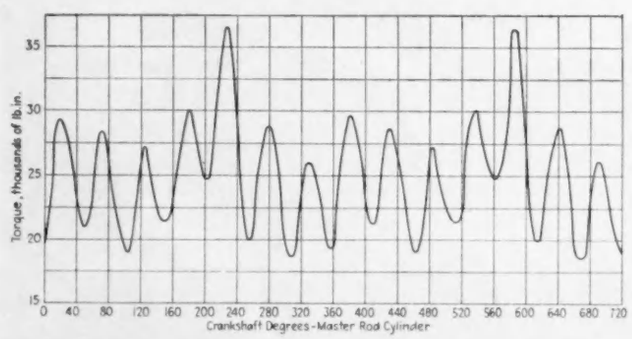


Fig. 13 – Typical Curve of Torque Variation Due to Gas Forces for Modern Fourteen-Cylinder Engine

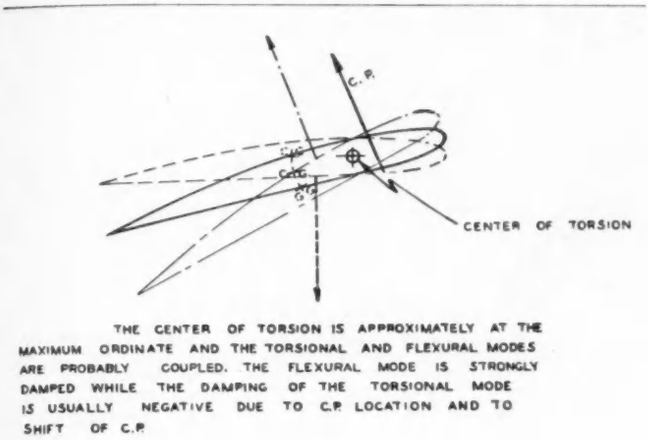

C.P.
CENTER OF TORSION
THE CENTER OF TORSION IS APPROXIMATELY AT THE MAXIMUM ORDINATE AND THE TORSIONAL AND FLEXURAL MODES ARE PROBABLY COUPLED. THE FLEXURAL MODE IS STRONGLY DAMPED WHILE THE DAMPING OF THE TORSIONAL MODE IS USUALLY NEGATIVE DUE TO C.P. LOCATION AND TO SHIFT OF C.P.

Fig. 15 – Coupling of Flexural and Torsional Oscillations

In Fig. 14 is indicated roughly the effect of aerodynamic flexural vibrations, showing that it is reasonable that a very considerable amount of damping is present.

In the case of the torsional vibration in the blade, however, the air forces are unstable so that the damping would be negative, and it is to be expected that stresses could reach very high values in case the torsional frequency is in resonance with the engine explosions. Actually, we have found this to be the case in our measurements and have found some extraordinarily high stresses under these conditions. Our practical experience also has indicated that propellers have comparatively low resistance to fluttering and that violent flutter usually leads to failure in a fairly short time. The coupling of flexural and torsional oscillations is illustrated in Fig. 15.

The form of the propeller blades is such that resonance vibrations, particularly in the fundamental and the first and second harmonics, produced very large linear movements of the blade element so that a very startling demonstration of the resonance vibrations may be obtained with a comparatively small amount of excitation provided the excitation is very closely held to the various vibration frequencies of the blade. See Figs. 16 and 17.

Quite a lot of work has been done on determining the natural periods of the non-rotating propeller and establishing the nodal points under these conditions.

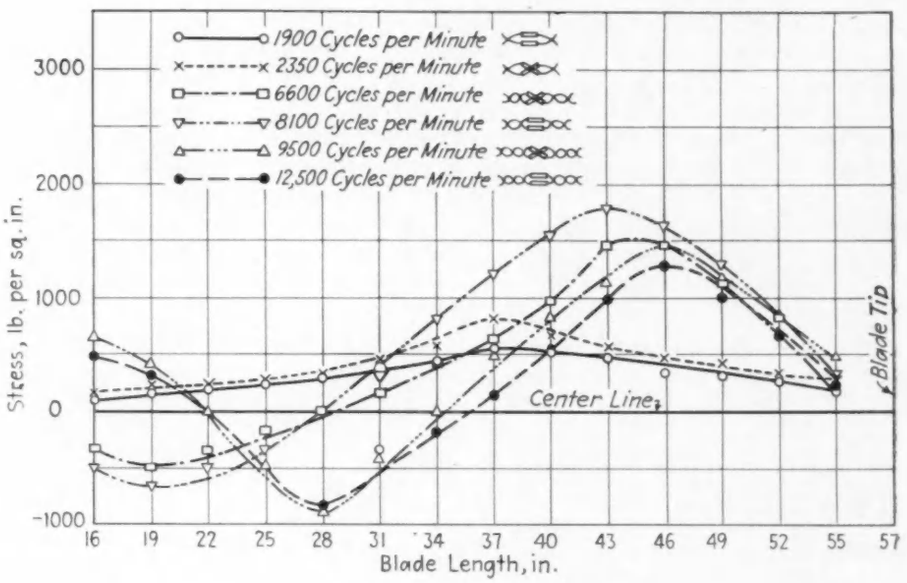


Fig. 16 (*Right*) – Example of Stress Distribution along the Blade for Various Modes of Resonance Vibrations

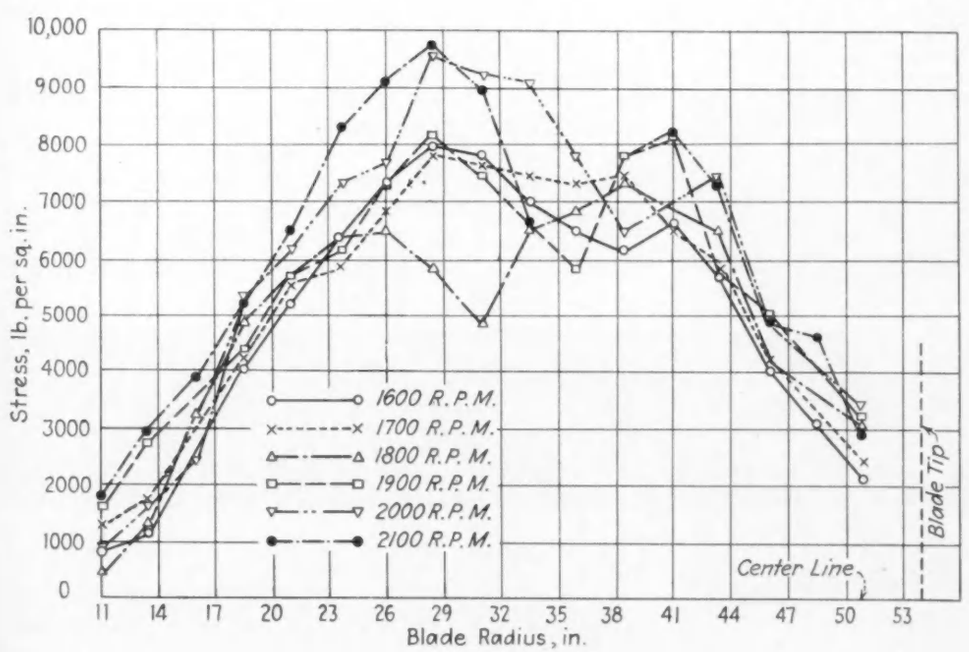


Fig. 17 (*Left*) – Example of Stress Distribution along the Blade when Running on the Engine, Indicating Presence of Forced Vibrations

Correction Factors for the Effect of Centrifugal Force on Various Types of Flexural Vibrations:

$$(f_R = \sqrt{f_o^2 + cN^2})$$

Type of Vibration	c (Theodorsen)	c (Hamilton Standard)
	1.7	1.5
	2.54*	-
	6.2	6.2
	-	-
	12.2	12.2
	8.73*	8.9

*These values are calculated by the method advanced by Theodorsen.

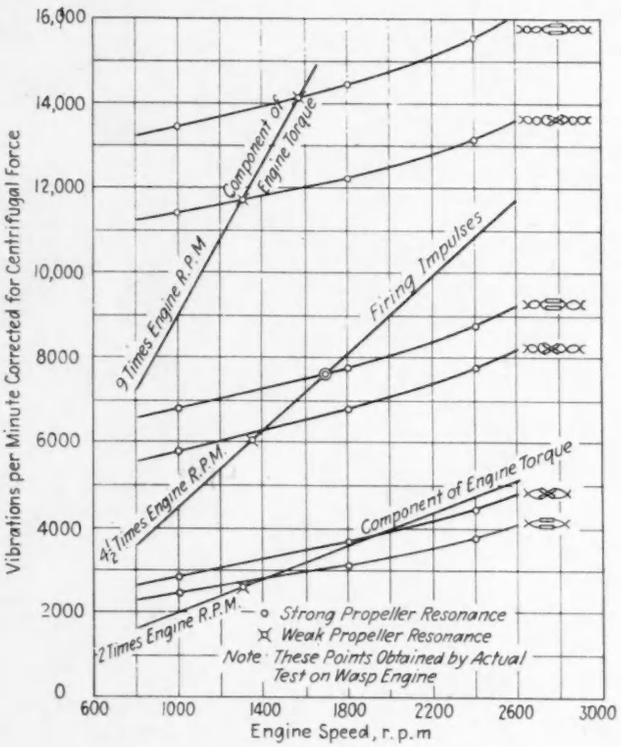


Fig. 18 – Types of Resonance Vibrations and Correction Factors for the Effect of Centrifugal Force – Statically Obtained Frequencies Corrected for Centrifugal-Force Effect

For the purpose of excitation, several schemes have been employed. At Wright Field a small device has been designed to produce rectilinear disturbances and this device is clamped to the hub of the propeller which is freely suspended in shock-absorber cords. For most purposes, almost the same result can be obtained with a rotating mass which is out of balance. Although this method produces accelerations along two axes, results obtained are quite similar to those obtained with rectilinear type of excitation.

At the National Bureau of Standards, a very ingenious scheme of mounting the propeller in the normal manner on the shaft of an electric motor and applying alternating current to the armature and direct current to the field is employed. By varying the frequency of the alternating current, excitation at various frequencies can be obtained and this excitation is applied in the same manner as the propeller shaft resonance in the case of the actual engine. In Dr. Lürenbaum's very excellent paper[2] presented before the Society last summer, the experiments at D.V.L. were described where a torsional excitation to the propeller shaft is applied by means of two unbalanced weights rotating in opposite directions at either end of a crank arm.

While the blades are being vibrated in the various forms, a stroboscopic light is employed to determine the nodal points and the phase relationship of the various blades. The phase relationship may be determined equally well by the use of electrical contacts held close to the vibrating tips. In-phase vibrations are indicated by a glowing lamp when both wires are touched to the cambered faces of the two tips; whereas out-phase vibrations are indicated by a glowing light when one wire is touched to the cambered side of one tip and the other wire, to the flat face of the other tip.

We have adopted the practice of attempting to correct the natural frequency with the propeller at rest for the effect of centrifugal force when the propeller is rotating by means of the following formula:

$$f_R = \sqrt{f_o^2 + cN^2}$$

where f_R = frequency of rotating propeller.

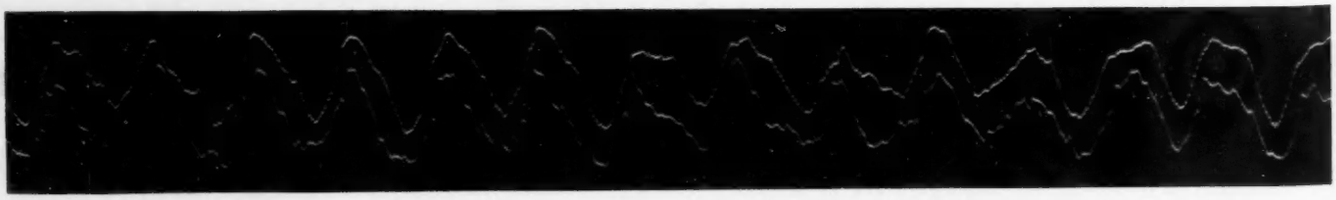
f_o = frequency of static propeller.

N = rate of rotation.

c = a numerical coefficient depending on the mode of vibration and the shape of the propeller blade.

Quite a number of experimental investigations of the value of the coefficient "c" have been undertaken, and very excellent experiments were carried out by Dr. T. Theodorsen, National Advisory Committee for Aeronautics, with blades of greatly reduced thickness and a set of coefficients established from these tests. In our own test work on the normal flight propellers we have found coefficients fairly close to those of Theodorsen. For purposes of convenient reference, I have tabu-

[2] See S.A.E. TRANSACTIONS, December, 1936, pp. 469-479; "Vibration of Crankshaft-Propeller Systems," by Karl Lürenbaum.



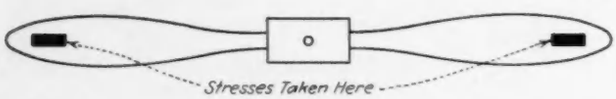


Fig. 19 – Example of Stress on Two Blades with Loop at Hub

This record is an example of stress at the stress loops near the tips of a two-way propeller which is vibrating with a node at the hub. The vibration frequency is 4½ times engine speed in resonance with cylinder explosions.

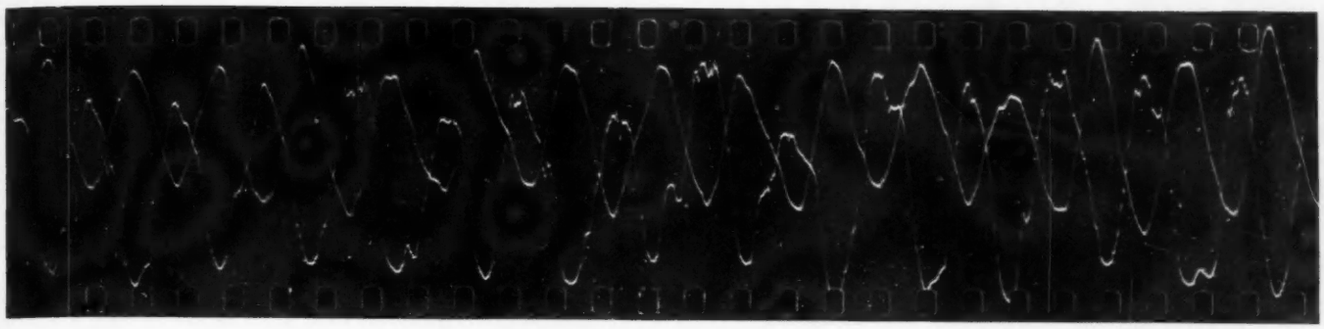
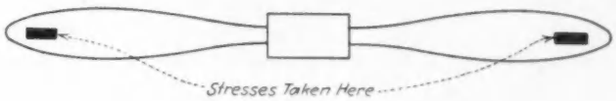


Fig. 20 – Example of Stress in Two Blades with Node at Hub

This record of two strain gages placed at vibrational nodes (stress loops) near the tips of a two-way propeller shows an example of stress with node at the hub. The exciting forces are at cylinder-explosion frequency of 4½ times engine speed.



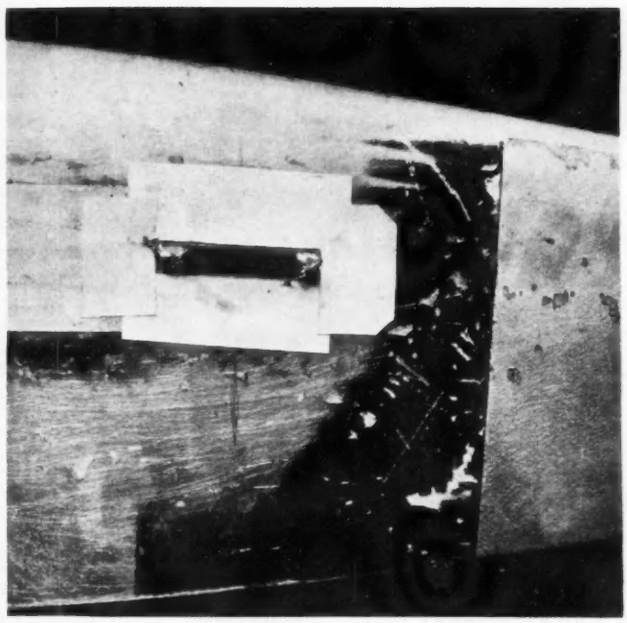


Fig. 21 – The Carbon-Resistor Type of Pickup with Insulation

lated in Fig. 18 the Theodorsen coefficients in comparison with the average value from our own experiments with actual propellers. See Figs. 19 and 20.

Methods of Measuring Stress

The problem of measuring the vibratory stresses of the running propeller has been approached in a number of different ways.

The greatest amount of effort has been applied to the use of strain gages and, among these, the types using electrical pickups appear to be the most adaptable. The electromagnetic strain gage has been used as a pickup but this type tends to become bulky, requires rather fine wires, and is necessarily somewhat fragile in construction. It has the further disadvantage of depending on velocity and requiring an integrating electrical circuit for the indications.

The scheme of inertia magnetic-type pickup also has been tried, but this method has the same disadvantages as the electromagnetic strain gage with the further disadvantage of being less positive in its action.

Some very good results have been obtained with the use of Piezo electric crystals, such as Rochelle salts, used as a pickup for the vibration stresses. This type of pickup measures curvature rather than stress, does not have a straight-line calibration of output against stress, and is affected by temperatures above 100 deg. fahr. The crystals also are quite fragile. Various apparatus and methods for measuring stress are illustrated in Figs. 21 and 22.

The best results obtained to date have been with a modified carbon-pile type of pickup in which a flat bar of carbon is cemented to the surface of the blade with a suitable insulator and the change in resistance, due to variation in stress, is measured by a suitable electrical circuit.

By using the proper binder for the carbon a very stable type of pickup not affected by ordinary variations in moisture and temperature has been obtained. The output is rather high so that stresses of a low order may be measured. The output also is proportional to the stress so that the problem of calibration is simplified. It has been found possible to calibrate the pickups under static loading, this calibration being checked and found unchanged under dynamic loading.

Quite a number of flight and engine tests already have been run with this set-up, and a great deal of valuable information has been obtained.

In addition to the electrical pickups, a good deal of work has been done with the scratch type of extensometer and some fairly successful records have been obtained.

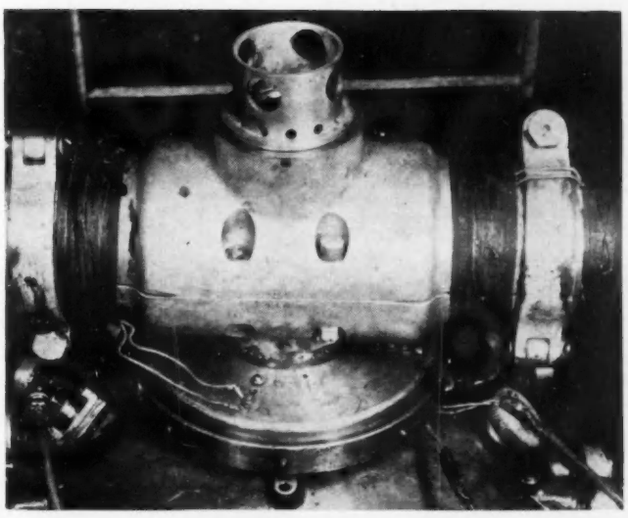


Fig. 22 – Slip-Ring Assembly

The work with mirror types employing a beam of light has only been partially successful.

Some information has been obtained from stroboscopic observations employing various types of stroboscopic lights such as the stroborama, stroboglow, and so on. However, it has been very difficult to get clear observations of the vibrations of the running propeller with the stroboscopic light, and the information obtained in this way has been very limited.

Some unusually interesting results have been obtained at Massachusetts Institute of Technology by the use of high-speed photography.

All of the various methods of testing stress measurement represent a useful effort, and it is likely that we shall have to continue to combine several kinds of tests in order to check the suitability of propellers from the standpoint of safe vibration stress.

Materials

Besides recognizing the stresses and sources of excitation in the propeller design, it is necessary as well to take into account the properties of the material being used.

The metals are, of course, particularly susceptible to weakening due to notch effects and, for this reason, it has been found necessary to proportion the propeller design in such a way as to avoid sudden change in section. Even though this factor is taken care of in the initial design, the propeller is exposed to very severe abrasion in service and a great many failures have been caused by the notch effect of sharp nicks. As a matter of interest, a tabulation is given in Fig. 23 showing the approximate effect on fatigue strength of steel, aluminum, and magnesium alloys, of the presence of notches of various proportions, as reported by J. B. Johnson, U. S. Army Air Corps.

The Probable Trend of Future Development

Up to the present time the propeller-design problem has been one of fitting a suitable propeller to an engine and airplane combination which had been established more or less independently of propeller characteristics.

Under these circumstances and in order to have a propeller design capable of reasonably extensive application, it has been necessary to attempt to build propellers of sufficient strength so that they would be capable of resisting very excessive vibratory excitations and so that they would not fail regardless of abuse in the way of applications and service operations.

The present trend toward very large powerplants in aircraft will lead to a very great increase in propeller weights if the present design practice is adhered to. These weights are apt to be so great that it will no longer be good practice to ask the propeller designer to fit a propeller to an independently developed engine and airplane combination, since the overall weight probably will be favored by a design which will permit favorable conditions for the propeller.

Whether the propeller for the large powerplants of the future will be one of very great diameter with a high reduction from crank speeds, a multiple-blade design of a somewhat higher rotational speed, or some type of counter-rotating propeller, will depend on a number of factors involving propeller efficiencies, weights, complications, the permissible clearances of aircraft structure, as well as considerations of vibratory stress.

Whichever line of development is followed, it appears very probable that it will be most advantageous to insulate the propeller from the engine as thoroughly as possible, since the types of excitation are so varied and of such severe magnitude that it appears almost impossible to design away from them in the propeller structure itself.

EFFECT OF GROOVE ON FORGED MAGNESIUM A.Z.N. ALLOY

ROTATING BEAM FATIGUE TESTS

		Unnotched									
		Room Temp.	−40°C.	Room Temp.	−40°C.	Room Temp.	−40°C.	Room Tempr.	−40°C.	Room Temp.	−40°C.
Fatigue Limit Notch / Fatigue Limit Unnotched	Ratio			.903	.833	.580	.666	.516	.556	.516	.556

EFFECT OF GROOVE ON FORGED 25S ALLOY

Fatigue Limit Notch / Fatigue Limit Unnotched	Ratio							.533	.444	.533	.444

EFFECT OF GROOVE ON − 6150 STEEL

		Standard									
End. Limit Notch / End. Limit	1		.382	.436	.364	.400	.455	.727	.655	.764	.816

EFFECT OF GROOVE ON − 25S ALLOY

End. Limit Notch / End. Limit	1		.554	.572	.572	.429	.821		.715	.92	

Fig. 23 – Tabulation of Effect of Notches on Properties of Steel, Aluminum and Magnesium Alloys